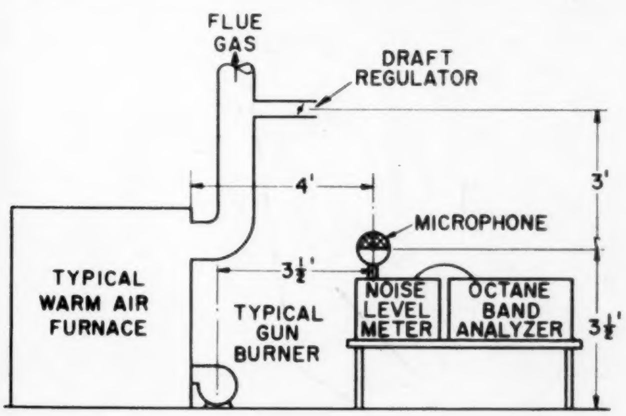
OFFICIAL PUBLICATION

ASHRAE JOURNAL

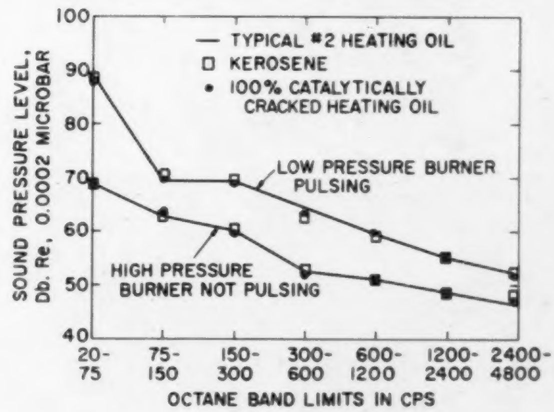
Heating • Refrigerating • Air Conditioning • Ventilating

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS

STANDARD EQUIPMENT USED FOR NOISE TESTS

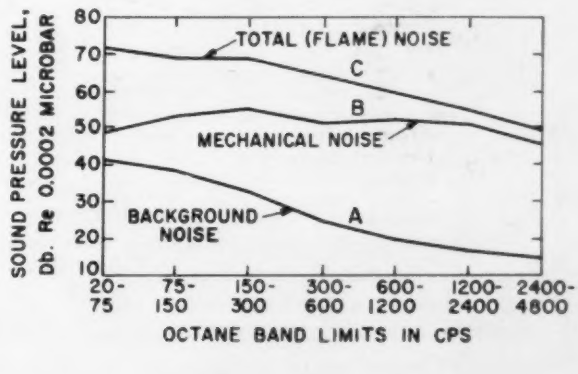


FUELS DO NOT INFLUENCE NOISE LEVELS



NOT FUELS BUT FLAMES PRODUCE MAJOR NOISE IS CONCLUSION OF OIL BURNER SOUND STUDY

THE FLAME IS THE MAJOR NOISE SOURCE



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Comment

REFORM OR NORM?

Something relatively scarce to recent times seems to be introducing itself quite broadly into the educational scene. Employers are reported to be getting highly choosey about the college graduates they interview and hire this year. One company in the fields of interest to ASHRAE had its recruiters talk to 3,000 college undergraduates before selecting 200 new employees from among them.

Educators may fuss and storm. Politicians may fume. But industry appears to be fed up with several things, according to the Wall Street Journal which prepared and reported upon this study. One of these things is mediocrity—just anybody will not do anymore—more employers now demand a man or woman from the top quarter of the graduating class. Another thing that industry would avoid is the floater—concentrated preliminaries are designed to eliminate the learn-and-run graduate. Recruiters, themselves, are no longer measured by their ability to bring in candidates; they have to produce a quality that links—with reasonable permanency—with the hiring organization.

Just where all this leads can be anybody's guess but ours is that repercussions will soon be thundering within the halls of ivy. For, if the product of the institution is unacceptable to the user of that product, something will have to give.

To our way of thinking, that's all for the good. It will improve the educational product—eventually—and improve the educational institution—inevitably. It will most assuredly have an ultimate effect on the grade school and the Deweyites who set up so much protective coloration these days that it is hard to recognize or pinpoint them when you do find them doing their hatchet job on the Three Rs and upon vestigial disciplines.

EDUCATION OR DISSEMINATION?

It is difficult to draw a fine line to separate the processes of educating formally and of supplying timely, pertinent facts. Be that as it may, we think that a professional engineering Society and its official publication are in the business of education.

This is no time to dodge responsibilities. On the contrary, it is a time to make the most of opportunities. All of which is just another way of saying that there can be no surcease in the fight for higher standards, better quality and more enlightened viewpoints in whatever we do.

Edward R. Searles
Editor

Late news highlights

API study

American Petroleum Institute is sponsoring a study to formulate an industry-wide research program to improve oil burner equipment and develop new uses for fuel oil. Already under way at Battelle Memorial Institute, the project will cover four distinct phases: survey of current research activities; evaluation of unconventional concepts; formulation of a research program; and stimulation and coordination of research.

Majority disapproves

Four out of five scientists and engineers working for large companies do not favor collective bargaining as a means of improving salary and social status, according to a University of Michigan survey. Professor John W. Riegen, director of the university's Bureau of Industrial Relations, stated the survey to reveal that nearly half of those opposed claimed collective bargaining was unnecessary and would offer no advantages. Other objections cited were that salaries and promotions would not reflect the individual's responsibilities and the reluctance of many to associate in a mass movement.

10th International Congress of Refrigeration

Appointment of six delegates to the 10th International Congress of the International Institute of Refrigeration, to be held in Copenhagen, Denmark, August 19-26, has been announced by Dr. Detlev W. Bronk, President of the National Academy of Sciences–National Research Council. The delegates, all members of the U.S. National Committee for IIR are: Dr. R. C. Jordan, J. E. Dube, W. T. Pentzer, F. G. Brickwedde, Dr. C. F. Kayan and B. H. Jennings. Included in the meeting will be a series of papers on the Peltier Effect, air conditioning and heat pumps, contributed from numerous countries.

Accidents reduced

An all time low in the frequency and severity of lost time accidents has been reached by members of the National Association of Refrigerated Warehouses. According to A. R. Carstensen, Chairman of the Safety Committee, lost time accidents for every million hours worked were reduced from 39½ in 1950 to 25 1/3 for 1958.

Milestone

At the 50th Annual Meeting of the National District Heating Association held June 1-4 at the Skytop Club, Skytop, Pa., officers elected were: George F. Prestwich, president; Albert F. Metzger, 1st vice president; John D. Lemon, 2nd vice president; and James C. Thompson, 3rd vice president. Erwin C. Bruce, Chairman of the Air Conditioning Committee, discussed "Air Conditioning Load" at this Committee's Report. During the Report of the Campus Heating Committee, with Thomas B. Kneen as Chairman, heating systems at Pennsylvania State University, Ohio State University and U. S. Air Force Academy were covered.

York says no

York Div of Borg-Warner Corporation will not join the Nema Room Air Conditioner Section on the basis that the Nema program does not achieve a "stabilization of ratings". States Austin Rising, vice president and director of marketing, "We feel that there is a need for a stronger program with firmer provisions for enforcement than now exists. . . . . In its present form the Nema program falls short of assuring distributors, dealers and customers of one satisfactory standard of Btu ratings within the room air conditioner industry."

Better business practices

Continuation of current improvements in appliance industry business practices will mean finer products, better service and expanded markets for all, B. A. Chapman, executive vice president and general manager of the Kelvinator appliance division of American Motors Corporation, told the 27th annual convention of the Institute of Appliance Manufacturers. Chapman decried "unwarranted and unrealistic expansion of production facilities" based simply on the desire for a larger share of the existing market. As a result, "the industry, in general, compensated for the discrepancy between virtually stable prices and rising costs by reducing service organizations, squeezing money out of long-term research for product improvement, and reducing advertising expenditures."

Residential heating A basic text on design, installation and operation of residential heating systems, **Winter Air Conditioning** by Seichi Konzo, J. Raymond Carroll and Harlan D. Bareither covers heating and ventilating in small buildings and homes. Working data, such as heat loss factors for various walls, design temperatures for numerous cities, degree-days and how to estimate fuel consumption, work sheets for heat loss calculations, radiator selection and duct sizing are included. This book is a companion volume to **Summer Air Conditioning** (RE, January 1959, p.20). The Industrial Press, 93 Worth Street, New York 13, N. Y., 630 pages, $8.

New gas refrigerator Norge Div of Borg-Warner Corporation is now testing a new gas refrigerator, Judson S. Sayre, president, has announced. If all goes well, the unit will be in mass production and marketed by mid-fall.

Air Pollution Conference At the National Conference on Air Pollution, Washington, D. C., November 18-20, extensive material related to air pollution control, much of it reported for the first time, was presented and discussed and recommendations were made which could point the way to action by governmental agencies, by industries and by private institutions and organizations. Proceedings of the 3-day Conference, sponsored by the U.S. Department of Health, Education and Welfare are available from the Superintendent of Documents, U.S. Government Printing Office, Washington 25, D. C., for $1.75 (526 pp.).

Airfoil heating "Study of Simplified Methods of Airfoil Heating" by T. R. Barksdale, J. N. Erazo and R. L. Fischer is an Air Force report which describes a simplified method of thermal anti-icing of aircraft using heater air, such as that available from the compressor section of a turbo-jet engine. The existing airfoil structure is used for heat distribution and only the addition of a distributor tube is required for installation of the system in an airfoil. Copies of the 82-page report (PB 151479) are available from OTS, U.S. Department of Commerce, Washington 25, D.C., $2.25.

Donation A gift of $1,250,000 from the Sarah Mellon Scaife Foundation for a new engineering building has been announced by Dr. John C. Warner, president of Carnegie Institute of Technology. To be known as the Alan M. Scaife Hall of Engineering, the new building will furnish space for expansion of graduate work and research in Physics and Metallurgical Engineering and provide more room for vital research in engineering materials. In addition, it will make possible increased enrollment in graduate studies.

Papers are published Proceedings of the Seventh Annual National Dairy Engineering Conference, Michigan State University, February 26 and 27, includes those 19 papers there presented. Copies are available for $2 from the University, East Lansing, Mich.

For prospective teachers In order to promote engineering education, the College of Engineering, University of Illinois, has established a program whereby ten selected seniors (or recent graduates) who are interested in becoming engineering teachers will be granted two-year internships in Mechanical Engineering and Electrical Engineering. To receive $2,000 each year for two years, the candidates will be permitted to pursue graduate studies towards master of science degrees. Special training under experienced teachers and seminars will be conducted as part of the program, which is sponsored by the Ford Foundation. Application should be addressed to: Professor S. Konzo, Coordinator of Teaching-Internship Program, Mechanical Engineering Building, Urbana, Ill.

Flexibility of pipes Because of serious hazards that can accompany plant or pipe failure, rule-of-thumb methods of stress evaluation are cited as no longer acceptable. **Analysis of Pipe Structures for Flexibility** by John Gascoyne has been prepared especially for the piping designer and covers the problems he encounters. The elastic-centre method of stress calculation is included as offering perhaps the best medium for explaining the phenomena encountered in thermal expansion problems. John Wiley & Sons, Inc., 440 Fourth Avenue, New York 16, N. Y. 181 pp., $7.50.

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Effect of fuel composition on

Oil burner noise

Noise is important to the purchaser of heating and air conditioning equipment, as has been amply demonstrated. This Society has devoted considerable effort towards understanding and solving the problems associated with noise. Much of this effort has been directed at the problems of mechanical noise such as compressor and blower operation, duct noise, insulation, etc. In oil-fired furnaces and boilers, however, much of the noise is nonmechanical in its origin and is, in fact, due to the flame itself. This noise is frequently referred to as combustion "roar."

There has been some feeling that fuel composition contributes heavily to the intensity of flame noise. Specifically, at the June, 1957, meeting of the ASHAE in Murray Bay, Quebec, there was considerable discussion of factors that influenced flame noise.* The discussions indicated divergent views on the effect of fuel composition, in this case No. 2 heating oil, in contributing to the amount of noise produced. Because data to answer this question were not readily available, an investigation was carried out to evaluate the effect of fuel composition on domestic oil burner noise.

Flame noise of two basic types is recognized. These are the normal combustion "roar" and occasionally a high amplitude "throbbing" known as pulsation. A variety of fuels were tested at both conditions using conventional burners.

* Discussion, ASHAE Transactions, 1957, p. 125-6.

William A. Beach, Richard W. Sage and Herbert F. Schroeder are with the Process Research Div, Esso Research and Engineering Company. This paper was presented at the ASHRAE annual meeting, Lake Placid, N. Y., June 22-24, 1959.

W. A. BEACH

R. W. SAGE

H. F. SCHROEDER

Typical furnace burner combination chosen – A conventional warm air furnace with an output rating of 72,000 Btu/hr was used for our test work. It is representative of the smaller sized furnace which is gaining popularity in new homes today. The oil burners tested in this furnace were also typical of those found in the field. A commercially available high pressure gun burner was used to study normal combustion noise; for pulsation flame noise, a typical low pressure air atomizing burner.

The combustion chamber was a 9-in. diam preformed chamber provided by the furnace manufacturer as an integral part of the unit. It was made of hard firebrick and is quite similar to the type installed in many modern heating plants.

All variables were held constant throughout the test except for fuel composition. The firing rate was adjusted for each fuel and burner to provide a constant heat input of 90,000 Btu/hr. This change was small, ranging from 0.65 to 0.76 gph for all fuels tested. The quantity of combustion air was adjusted for each case to be 30% in excess of stoichiometric requirements. With No. 2 heating oil this was the setting required for zero smoke. For some of the other fuels, however, this setting caused considerable smoke formation. Draft was maintained at 0.015 in H_2O measured over the fire.

Accurate instruments used to examine entire sound spectrum – Burner noise is composed of sound waves of all frequencies in the audible spectrum (20-20,000 cps). However, in most test work, it is not necessary to analyze the spectrum for sound pressure level as a continuous function of frequency, rather, the sound level is measured in "octave bands." These octave bands have the same frequency limits as octaves on the piano keyboard. The sound pressure level gradient obtained by this method gives a good approximation of the entire spectrum and was used in this analysis of burner noise.

The sound instruments used in the program gave results reproducible within 1 decibel. Under normal conditions the human ear can barely detect this one decibel change in sound pressure level. A commercially available sound level meter was used for detection, and sound measurements in the octave

bands were made with a standard octave band frequency analyzer. These instruments were checked and calibrated daily. They were always located in the same position relative to the furnace and this location was chosen as the point at which the sound pressure level was most critical. The microphone orientation is shown in Fig. 1.

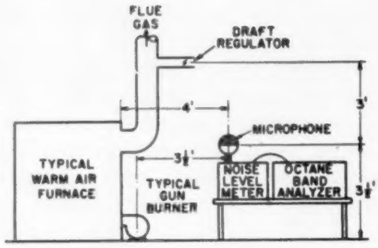
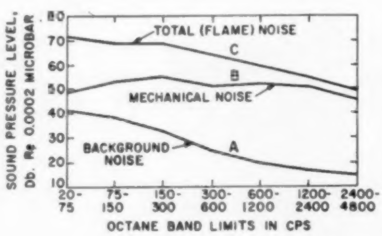


Fig. 1 Microphone orientation used in tests here reported upon

Sound measurement revealed the flame to be the major noise source – Sound measurements of the possible oil burner noise sources showed that only the flame contributed significantly to the total noise level. Three noise sources were considered, (1) background or extraneous noise levels, (2) mechanical noise from the burner and furnace, and (3) the flame itself. Fig. 2 shows the cumulative noise levels built up from these three components.

Fig. 2 Cumulative noise levels built up from three components



The background noise level, shown as Curve A, was measured with the burner and furnace turned off. Since the sound laboratory was remotely located, it had an extremely low background noise level, the equivalent of a radio broadcasting studio. The sound pressure level (SPL) gradient obtained was at least 30 db below the total burner noise in all octave bands. Since the background noise level was so much less than the burner noise it did not contribute to the total noise level. This is due to the logarithmic basis of the decibel sound level scale which means that decibel readings cannot be added arithmetically. For example, if two noise sources have individual sound levels of 80 db, the total sound level when running together would be 83 and not 160 db. As the individual sound levels become further and further apart, the combined level becomes essentially that of the higher level noise source.

The mechanical noise level shown as Curve B of Fig. 2 was measured with all the mechanical components of the burner and furnace operating. This noise level included the furnace blower, oil pump, motor, air fan and ignition system. The mechanical noise level was also sufficiently below the total burner noise level to render it a minor noise contributor. Since the only other noise source was the flame itself, it is obvious that the flame was the major source of noise in this oil burner. Curve C of Fig. 2 therefore represents both the total noise level and the flame noise level.

Fuel composition does not affect flame noise – Nine different fuels were tested for their effect on the largest oil burner noise contributor, the flame. These test fuels included typical products now marketed, fuels composed entirely of cracked products and several pure compounds. The properties of these fuels are listed in Table I.

Despite wide variations in fuel composition, there were no significant differences in the overall noise level or in the noise level in any of the individual octave bands with fuels boiling in the range of kerosenes and No. 2 home heating oils. A fuel composed of 100% catalytically cracked oil and kerosene, a straight run virgin product, gave the same sound spectrum as regular No. 2 heating oil in both the low pressure and high pressure gun burners. These SPL gradients are shown in accompanying Fig. 3.

The results were independent of the type burner used. Although the high pressure burner SPL gradient is inherently lower than the low pressure burner SPL gradient, it is, nevertheless, similarly insensitive to fuel changes. Even the highly unsaturated methyl dicyclopentadiene, which can be considered an extreme of possible cracked oils, did not vary from the noise level of No. 2 heating oil. Normal

(*Continued on page 68*)

TABLE I

SPECIFICATIONS OF FUELS TESTED

	Gravity	Composition, Wt. %				Boiling Range [1] F		
	°API	Paraffin	Naphthene	Olefin	Aromatic	10%	50%	90%
No. 2 Heating Oil	35.5	37.1	29.9	2.3	30.7	396	478	562
Kerosene	41.2	39.6	41.6	0.6	18.2	382	434	487
100% Cat. Cracked Heating Oil	30.0	——53.4——		12.6	34.0	355[2]	465	610[2]
100% Thermally Cracked Heating Oil	36.4	——37.7——		28.8	33.5	330[2]	442	590[2]
Methyl Dicyclopentadiene	19.3	0.0	0.0	100.0	0.0	330	330	330
Benzene	28.6	0.0	0.0	0.0	100.0	176	176	176
n-Cetane	51.1	100.0	0.0	0.0	0.0	550	550	550
n-Heptane	74.2	100.0	0.0	0.0	0.0	209	209	209
Isooctane	71.8	100.0	0.0	0.0	0.0	211	211	211

[1] Appropriate ASTM Distillations.
[2] Initial and final boiling points.

Digital computer applied to

Compressor design analysis

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Forces acting on the piston, cylinder, piston pin, connecting rod, crankpin and crankshaft power train must be known to design rationally various components of reciprocating machines. These forces are caused by gas pressure in the cylinders and inertia forces of the moving parts. The gas pressure forces and oscillating inertia forces change in magnitude and direction as the crank rotates. The rotating inertia forces have a constant magnitude but change direction as the crank rotates.

The computation of the resultant forces acting upon each component of the power train at successive crank intervals is a long and tedious procedure, even in the simplest case of a single-cylinder compressor operating at a given suction and discharge pressure. As an order of magnitude 16 engineering manhours are required to complete the force analysis of a single-cylinder compressor at 15-deg crank angle intervals for one complete revolution of the crank. For an 8-cyl compressor design with a 2-throw crankshaft, the estimated time for a similar solution is 30 engineering manhours of repetitive mathematical and graphical work. In normal compressor design it is necessary to determine the resultant forces, in direction and magnitude, under several operating conditions. The time element may become so prohibitive that the designer may not even attempt to compute the forces acting on the various components and instead resort to rough rules of thumb. These short cuts are not justifiable (and in some instances quite risky) in a competitive market requiring closely engineered yet reliable and durable products.

Primary purpose of this project was to set up a general calculation program for the power train forces using a digital, card-programmed computer (CPC). The computer takes over the laborious repetitive calculation with electronic speed and accuracy once the program has been set up. For comparison with the manual computation time cited in the previous paragraph, the IBM computer will complete the calculation of a single-cyl compressor in approximately ¾ hr, or 5% of manual computation time, and the 8-cyl, 2-throw compressor solution in 6 hr or 20% of manual computation time. The total machine running cost, including burden, will depend on the computer usage. In this application it is approximately $18 per hr. Assuming an average cost of $6 per engineering manhour, or in this case one-third of the computer hourly cost, the economical computer break even point is reached if the machine time is equal or less than one-third of the manual computation time. With a machine to manual time ratio of 5% to 20%, the application of the CPC to the solution of single and multi-cylinder compressor force calculation obviously is justified.

SINGLE-CYLINDER COMPRESSOR CALCULATION

1. Gas Pressure Forces

The gas forces are computed from the pressure-volume diagram shown in Fig. 1.

The cylinder volume at a crank-angle, (α) with clearance factor (C) is expressed by

$$v = \left[r(1 - \cos\alpha + \frac{\lambda}{2} \sin^2\alpha) + 2rC \right] A_p \qquad (1)$$

and with

$$(2C + 1 - \cos\alpha + \frac{\lambda}{2} \sin^2\alpha) = x,$$

$$v = r\,x\,A_p \qquad (2)$$

H. Soumerai is Chief Engineer, Refrigeration Compressor Engineering, T. Kusuda is Research Engineer, both with the Air Conditioning and Refrigeration Div, Worthington Corporation. Professor J. H. Dittfach is in charge of the Heat Power Laboratories at the University of Massachusetts. This article is a condensation of the paper presented at the 45th Semiannual Meeting of ASRE in New Orleans, La., Dec. 1-3, 1958, as "Reciprocating Gas Compressor Forces—Calculation on the Card Program Computer". For complete details of the computer program, sample calculations, tabulations and figures omitted in this condensation, contact Dr. T. Kusuda or H. Soumerai, Worthington Corporation, Holyoke, Mass.

During the expansion stroke,

$$P_d v_1^n = P_s v_2^n = P v^n \tag{3}$$

And with the net cylinder pressure during the expansion stroke will be

$$P - P_s = P_d\left(\frac{2C}{x}\right)^n - P_s = P_g \tag{4}$$

In the same manner, during the compression stroke,

$$P_d v_4^{\frac{1}{n}} = P_s v_3^{\frac{1}{n}} = P v^{\frac{1}{n}} \tag{5}$$

Then,

$$P - P_s = P_s \frac{2C + 2}{x}^{n^1} - P_s = P_g \tag{6}$$

For the remaining portions of the cycle, constant suction and discharge pressures are assumed (valve throttling neglected);
$P_g = 0$ during the suction stroke, and $P_g = P_d - P_s$ during the discharge stroke.
The gas pressure force is the product of P_g and A_p.

2. Connecting Rod Forces
The acceleration of the piston can be written as

$$\omega^2 r\ (\cos\alpha + \lambda\cos 2\alpha) = \omega^2 r Z \tag{7}$$

The total reciprocating mass is the sum of the piston assembly mass and a part of the connecting rod mass,

$$\frac{W_{RE}}{g} = \frac{W_P}{g} + \frac{W'_{RE}}{g} \tag{8}$$

The usual practice (Ref. No. 5) is to divide the connecting rod weight into reciprocating (W'_{RE}) and rotating portions (W_{RO}) proportional to distances from the crankpin to the center of gravity and from the piston pin to the center of gravity respectively, or:

$$W'_{RE} = W_c \frac{d}{l} \tag{9}$$

The total reciprocating inertia force will be,

$$F_i = -\frac{W_{RE}}{12\,g}\omega^2 r Z \tag{10}$$

Since

$$\omega = \frac{2\pi N}{60} = \frac{\pi N}{30}, \text{ and } g = 32.2 \text{ ft/sec}^2$$

equation (10) becomes

$$F_i = -2.84 \cdot W_{RE}\, r N^2 Z \cdot 10^{-5}\ \text{(lb)}$$

Or, with

$$F_{RE} = -2.84\, W_{RE}\, r\, N^2 \times 10^{-5}$$
$$F_i = F_{RE} Z \tag{11}$$

The sign convention can be simply explained by stating that all forces acting toward the center of the main crankshaft axis are positive (rod in compression). Since at the beginning of the expansion stroke Z is positive and the rod is in tension, F_{RE} has the negative sign.

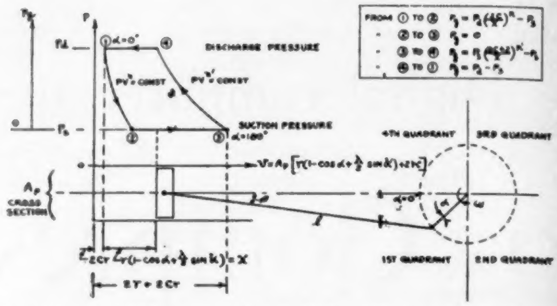


Fig. 1 P-V Diagram of a polytropic gas compressor

The total force transmitted along the connecting rod axis will be, from Fig. 2A

$$F_c = \frac{A_p P_g + F_i}{\cos\beta} = \frac{A_p P_g + F_i}{\sqrt{1 - \lambda^2 \sin^2\alpha}} \tag{12}$$*

Since

$$\sin\beta = \lambda \sin\alpha$$

The radial inertia force of the rotating portion of the connecting rod acting on the crankpin is

$$F_{RO} = -\frac{W_{RO}}{12\,g} \cdot r\,\omega^2$$

or

$$F_{RO} = -2.84\, W_{RO}\, r\, N^2 \times 10^{-5} \tag{13}$$

where

$$W_{RO} = \frac{W_c}{l}(l - d) \tag{14}$$

The two connecting rod forces F_C and F_{RO} are combined after resolving F_C into F_{CX} and F_{CY} components as follows: (See Fig. 2B)

$$F_{cpy} = F_c \sin(\alpha + \beta) = F_c\ (\sin\alpha \sqrt{1 - \lambda^2 \sin^2\alpha} + \lambda \sin\alpha\cos\alpha) \tag{15}$$

This tangential force acting on the crankpin is positive in the direction of rotation.
And the radial component on the crankpin is

$$F_{cpx} = F_c\ (\cos\alpha \sqrt{1 - \lambda^2 \sin^2\alpha} - \lambda \sin^2\alpha) + F_{RO} \tag{16}$$

This force is positive in the centripetal direction. Combining these two forces into a resultant force on the crankpin gives,

$$F_{cp} = \sqrt{F^2_{cpx} + F^2_{cpy}} \tag{17}$$

3. Forces on Main Crankshaft Bearings
The rotating masses of the crankshaft and connecting rod system can always be completely balanced by counterweights (Ref. No. 6). Therefore it is not necessary to consider the rotating inertia

* Note 1. Note that expression (12) can also be used to compute the wrist pin load; however, in this latter case the term W'_{RE} in equation (8) is dropped out and $W_{RE} = W_P$ piston assembly weight including or excluding wrist pin weight if wrist pin is clamped in the piston or rod respectively.

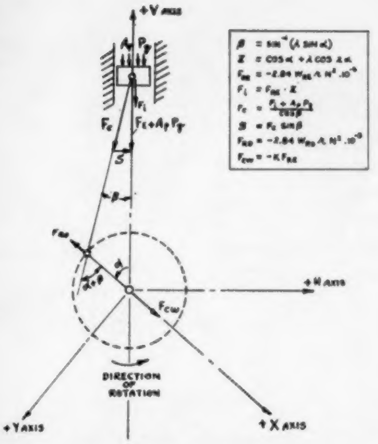


Fig. 2A Single-cylinder reciprocating compressor forces and their locations

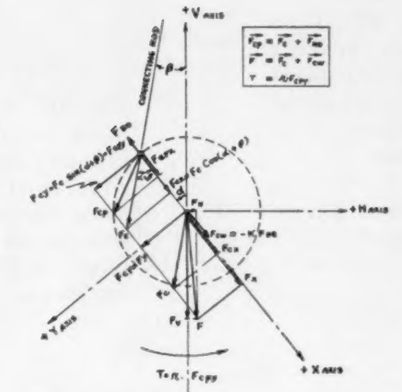


Fig. 2B Single-cylinder reciprocating compressor, force vector diagram

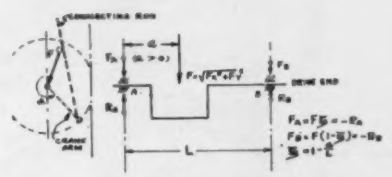
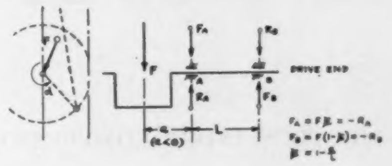


Fig. 3A Single throw crankshaft forces with center crank throw

Fig. 3B Single throw crankshaft forces with overhung crank throw



forces to compute the main bearing reactions in a modern dynamically balanced reciprocating machine. The only active forces contributing to the main bearing reactions are the connecting rod force, F_C, and the resultant reciprocating counterbalance force F_{CW} which is applied to partially offset the oscillating inertia forces of the piston assembly and reciprocating portion of the connecting rod. The resultant reciprocating counterweight can be visualized as a rotating weight $W_{CW} = K \times W_{RE}$ concentrated at a distance, r, from the crankshaft axis in the same plane as the connecting rod force F_C (See Fig. 2B), thus

$$F_{cw} = +2.84\, K\, W_{RE}\, r\, N^2 \times 10^{-6} \qquad (18)$$

where

$$K \leqq 1, \text{ or } F_{cw} = -K\, F_{RE} \qquad (19)$$

The fraction K (frequently 0.5) is so selected as to minimize the effect of reciprocating inertia forces. In practice the resultant counterweight KW_{RE} is replaced by an equivalent pair of weights located in any convenient plane along the crankshaft axis. Rotating and reciprocating counterweights are usually physically combined in actual compressor design; however, the mathematical analysis is simplified considerably by treating the reciprocating counterweight as a separate entity. The two coplanar forces F_c and F_{cw} are reduced to a force, F, acting on the main crankshaft and a torque T, as shown in Fig. 2B.

$$F = \sqrt{F_x^2 + F_y^2} \qquad (20)$$

where

$$F_x = F_{cx} + F_{cw} \qquad (21)$$
$$= F_c\,(\cos\alpha\,\sqrt{1-\lambda^2\sin^2\alpha} - \lambda\sin^2\alpha) + F_{cw}$$
$$F_y = F_{cpy} = F_{cy} \qquad (22)$$
$$= F_c\,(\sin\alpha\,\sqrt{1-\lambda^2\sin^2\alpha} + \lambda\sin\alpha\cos\alpha)$$

and

$$T = r\,F_y = \text{Instantaneous torque} \qquad (23)$$

To distribute the resultant force, F, on main bearings A and B, the crankshaft is treated as a simply supported beam** (Ref. 1), then

$$F_A = F\xi \qquad (24)$$
$$F_{AX} = F_X\xi$$
$$F_{AY} = F_Y\xi$$

and

$$F_B = F\,(1-\xi)$$
$$F_{BX} = F_X\,(1-\xi) \qquad (25)$$
$$F_{BY} = F_Y\,(1-\xi) \qquad (26)$$

where

$$\xi = 1 - \frac{a}{L}$$

If it is desired to compute the force components with reference to fixed coordinates V and H, the following equations are to be used:

$$F_H = F_X \sin\alpha - F_Y \cos\alpha \qquad (27)$$
$$F_V = -F_X \cos\alpha - F_Y \sin\alpha \qquad (28)$$

with the sign convention shown in Fig. 2.

The calculated forces on the crankpin and main bearings are very conveniently represented by vectorial plots (See Figs 4A and 4B). These diagrams are easily made using the X and Y components. In plotting the data, careful attention must be given to the proper vectorial layout and sign conventions. As shown in Fig. 3 the actual bearing reaction forces from bearing to journal, R_A and R_B, have the same value as F_A and F_B but opposite directions. Note that in the special case of an overhung crank such as shown in Fig. 3B, the negative value is to be used for a.

** In some cases the crankshaft may not be treated as a simply supported beam. An excellent treatise on this subject is available in Reference (7) for a single throw between the two bearings. We plan to discuss this subject for double throws between two supports in a future article.

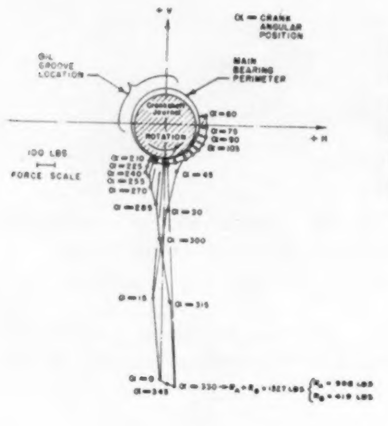


Fig. 4A Total main bearing reaction force $R_A + R_B$; single-cylinder compressor

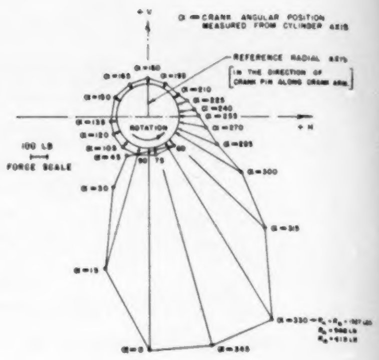


Fig. 4B Total main bearing journal force distribution along the journal perimeter; single - cylinder compressor

MULTI-CYLINDER COMPRESSOR CALCULATION

In this section the basic single-cylinder equations are extended to a multi-cylinder compressor design. The computer program is designed to handle the more general double throw machine; however, the program can be applied to the special case of a single throw compressor calculation by introducing the proper constants in the expanded program.

In order to make the calculation as simple as practicable for actual compressor design applications, the following assumptions are made:

1. The generalized multi-cylinder machine has a scheme of cylinders and cranks such as is shown in Fig. 5.
   a. Equal number of cylinders and identical arrangement on each crank.
   b. Cylinder rod assemblies are identical.
   c. Compression ratio is identical for all cylinders, (single stage operation).
2. For each crank throw all connecting rod gas and inertia forces and the resultant reciprocating counterweight forces are acting in a common plane perpendicular to the crankshaft axis and located axially in the center of the crankpin.
3. All the rotating forces of the crank assembly are perfectly counterbalanced.
4. The crankshaft is treated as a simply supported beam with two concentrated loads applied at transverse centerline position of each crankpin.

No limitations are placed on the number of cylinders per throw, m; angle between two consecutive cylinders, $\Delta\gamma$; angle between the two cranks, ε; crankpin transverse centerline locations, $a^{(1)}$, $a^{(2)}$ and all other design parameters.

The force analysis is now made on the first throw for the vth cylinder from the reference cylinder. The relative crank angle $\alpha_v^{(1)}$ becomes

$$\alpha_v^{(1)} = \alpha_1^{(1)} + (v - 1)\,\Delta\gamma \qquad (29)$$

where $\alpha_1^{(1)}$ is the angle between the first throw crank and the reference cylinder axis in the direction of rotation (counter-clockwise on Fig. 7).

The numbering system of cylinders and crank throws is shown on Fig. 5. Similarly the second throw crank angle for the vth cylinder is

$$\alpha_v^{(2)} = \alpha_1^{(2)} + (v - 1)\,\Delta\gamma \qquad (30)$$

$$\alpha_1^{(2)} = \alpha_1^{(1)} + \varepsilon \qquad (31)$$

Rewriting equation (12), the connecting rod forces of the vth cylinder on the first and second throws are respectively

$$F_{cv}^{(1)} = \frac{A_p P_{gv}^{(1)} + F_{Iv}^{(1)}}{\sqrt{1 - \lambda^2 \sin^2 \alpha_v^{(1)}}} \qquad (32)$$

and

$$F_{cv}^{(2)} = \frac{A_p P_{gv}^{(2)} + F_{Iv}^{(2)}}{\sqrt{1 - \lambda^2 \sin^2 \alpha_v^{(2)}}} \qquad (33)$$

The crankpin forces on the first throw can be expressed in the first throw $(X^{(1)}Y^{(1)})$ coordinate system as follows—

$$F_{cpy}^{(1)} = \sum_{v=1}^{m} F_{cv}^{(1)} [\sin \alpha_v^{(1)} \sqrt{1 - \lambda^2 \sin^2 \alpha_v^{(1)}} + \lambda \sin \alpha_v^{(1)} \cos \alpha_v^{(1)}] \qquad (34)$$

$$F_{cpx}^{(1)} = \sum_{v=1}^{m} F_{cv}^{(1)} [\cos \alpha_v^{(4)} \sqrt{1 - \lambda^2 \sin^2 \alpha_v^{(1)}} - \lambda \sin^2 \alpha_v^{(4)}] + m F_{RO} \qquad (35)$$

and

$$F_{cp}^{(1)} = \sqrt{(F_{cpx}^{(1)})^2 + (F_{cpy}^{(1)})^2} \qquad (36)$$

The crankpin forces on the second throw $F_{cpy}^{(2)}$, $F_{cpx}^{(2)}$ and $F_{cp}^{(2)}$ can be expressed in the same manner in the second throw $(X^{(2)}Y^{(2)})$ coordinate system by substituting superscript (2) for (1) in the above equations.

As in the case of the single cylinder compressor, perfect balance of all rotating inertia force is assumed and the forces on the main bearing A exerted by the first throw elements are expressed as follows:

$$F_{AY}^{(1)} = \sum_{v=1}^{m} F_{cv}^{(1)} [\sin \alpha_v^{(1)} \sqrt{1 - \lambda^2 \sin^2 \alpha_v^{(1)}} + \lambda \sin \alpha_v^{(1)} \cos \alpha_v^{(1)}]\, \xi^{(1)} \qquad (37)$$

$$F_{AX}^{(1)} = \sum_{v=1}^{m} F_{cv}^{(1)} [\cos \alpha_v^{(1)} \sqrt{1 - \lambda^2 \sin^2 \alpha_v^{(1)}} - \lambda \sin \alpha_v^{(1)}]\, \xi^{(1)} + m F_{cw}\, \xi^{(1)} \qquad (38)$$

$$\xi^{(1)} = 1 - \frac{a^{(1)}}{L}$$

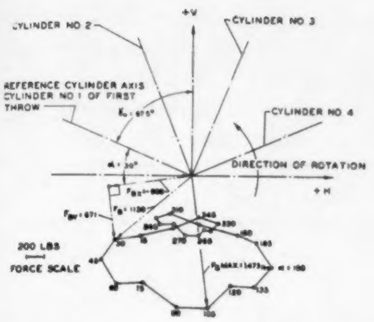


Fig. 5 Generalized multi-cylinder double throw compressor scheme

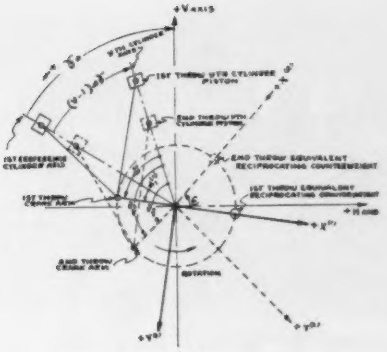


Fig. 6 Main bearing Force Polar Diagram F_B, 8-cylinder double throw compressor

The forces on the main bearing B exerted by the first throw elements: $F_{BY}^{(1)}$ and $F_{BX}^{(1)}$, are expressed in the same manner as $F_{AY}^{(1)}$ and $F_{AX}^{(1)}$ respectively with the exception that $1-\xi^{(1)}$ is substituted for $\xi^{(1)}$.

Main bearing forces exerted by the second throw elements, $F_{AY}^{(2)}$, $F_{AX}^{(2)}$, $F_{BY}^{(2)}$, and $F_{BX}^{(2)}$ have the same expression in the $(X^{(2)}Y^{(2)})$ coordinate system as $F_{AY}^{(1)}$, $F_{AX}^{(1)}$, $F_{BY}^{(1)}$ and $F_{BX}^{(1)}$ in the $(X^{(1)}Y^{(1)})$ system with the superscript (1) being replaced by (2).

In order to add vectorially all the main bearing forces, the rotating coordinates $(X^{(1)}Y^{(1)})$ of the first throw are taken as the system reference coordinates (See Fig. 7).

Then

$$F_{AY} = F_{AY}^{(1)} + F_{AY}^{(2)} \cos \varepsilon - F_{AX}^{(2)} \sin \varepsilon \qquad (39)$$

$$F_{AX} = F_{AX}^{(1)} + F_{AX}^{(2)} \cos \varepsilon + F_{AY}^{(2)} \sin \varepsilon \qquad (40)$$

and F_{BY}, F_{BX} have the same expression as (39) and (40) respectively with B substituted for A. Thus we have finally the resultant main bearing forces

$$F_A = \sqrt{F_{AX}^2 + F_{AY}^2} \qquad (41)$$

$$F_B = \sqrt{F_{BX}^2 + F_{BY}^2} \qquad (42)$$

In order to handle both single and double throw compressor force calculations on the same generalized computer program, equations (39) and (40) are rewritten in the following manner.

$$F_{AY} = \sum_{k=1}^{k} \left[F_{AY}^{(k)} \cos \varepsilon^{(k)} - F_{AX}^{(k)} \sin \varepsilon^{(k)}\right] \qquad (43)$$

$$F_{AX} = \sum_{k=1}^{k} \left[F_{AX}^{(k)} \cos \varepsilon^{(k)} + F_{AY}^{(k)} \sin \varepsilon^{(k)}\right] \qquad (44)$$

In the case of a double throw crankshaft k = 2

$$\varepsilon^{(1)} = 0$$

$$\varepsilon^{(2)} = \varepsilon \neq 0 \qquad (45)$$

and in the special case of a single throw crankshaft k = 1

$$\varepsilon^{(1)} = 0$$

$$\varepsilon^{(2)} = \varepsilon = 0$$

In some applications it is desirable to use a fixed reference coordinate system (V, H) for F_A and F_B. Then

$$F_{AH} = F_{AX} \sin (\alpha_1 + \gamma_o) - F_{AY} \cos (\alpha_1 + \gamma_o) \qquad (46)$$

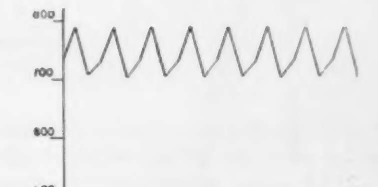


Fig. 7 8-cylinder double throw compressor torque diagram

$$F_{AV} = - F_{AX} \cos (\alpha_1 + \gamma_o) - F_{AY} \sin (\alpha_1 + \gamma_o) \qquad (47)$$

Where γ_o is the angle between the reference cylinder axis and the vertical fixed axis V as shown in Fig. 5.

The total crankshaft torque T is the algebraic sum of the first and second throw torque, or

$$T = r\,(F_{cpy}^{(1)} + F_{cpy}^{(2)}) \qquad (48)$$

Figs. 6 and 7 illustrate the computer results of F_B and T 8-cylinder, double throw, Refrigerant-22 compressor.

In this particular case the main bearing force diagram F_A has the same shape as F_B except that the angular marking is shifted 180 deg. The reason for this is that the crankshaft under consideration is constructed symmetrically along the main shaft axis $(a^{(1)} = a^{(2)})$.

DISCUSSION AND LIMITATIONS OF THE PROGRAM

Several assumptions were made to simplify the mathematical treatment and to keep the machine computation time within economical limits. These assumptions impose some limitations on the validity of the program; these limitations are:

1. In line with the current design practice the reciprocating inertia forces of third (3α) and higher order have been neglected and the connecting rod reduced to

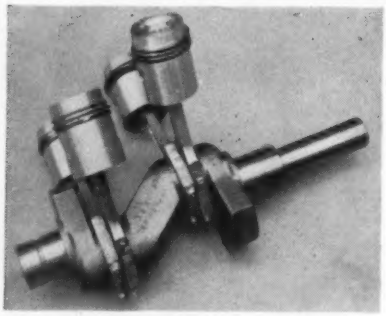


Fig. 8 Modern compressor crankshaft and rod assembly, 4-cylinder double throw

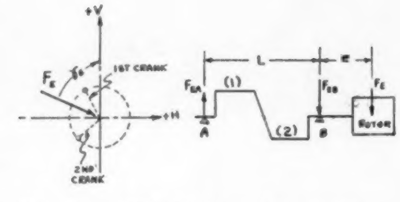


Fig. 9A Effect of random motor pull on main bearing force

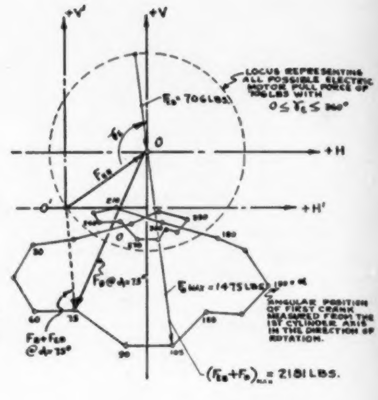


Fig. 9B Polar diagram of forces on main bearing B (F_B and F_{EB})

a reciprocating weight concentrated at the wristpin and a rotating weight concentrated at the crankpin.

2. All static weight and friction forces are neglected. Actually, the static weight and frictional forces have little effect on the compressor force calculation in modern force-feed lubricated machines operating near or above design load conditions.

3. The compression cycle employed assumes polytropic expansion and compression (exponents n and n^1) and zero valve losses during suction and discharge strokes. If desired, the effect of valve pressure drop at both pressure levels can be included in the program.

4. The rotating inertia masses of the crankshaft-connecting rod system are perfectly counterbalanced. The only inertia forces contributing to the main bearing reactions are the reciprocating inertia forces and "reciprocating" counterweight forces (counterweight factor: K).

5. The crankshaft is treated as a rigid member freely supported on its two main bearings.

6. The program does not apply to a system with three (or more) main bearings. However, the program could be extended to include this type of design if necessary. To do so would require the use of the theorem of three moments, treating the crankshaft as a beam with concentrated variable loads. This extended program could also consider the effect of bearing misalignments.

In applying the CPC computer the final program was tailored specifically to modern high speed single stage compressors. This justifies the following additional assumptions and limitations.

7. The program is set to handle a maximum of two throws. The unusual three-throw design can be treated as the combination of a two-throw and single-throw compressor. The main bearing forces are obtained by superposition of the computed results for the two-throw and one-throw compressor.

8. The program can only handle single stage machines, i.e., all cylinders are identical and operate at the same suction and discharge pressure. However, multistage compressors can be handled by graphical or algebraic superposition.

9. The assumption was made that all the connecting rod forces in each crank act in a common transverse plane through the centerline of the crankpin. This assumption is justified for most modern multicylinder compressors which have a relatively short crankpin length compared with the distance between crankpin and main bearings. (Fig. 8). In those cases where this assumption is not justified, it is possible to let each connecting rod force act in its true plane and determine the main bearing force by superposition.

10. No external forces such as belt or electric motor magnetic pull are included in the program. These forces can simply be added to the computer results by arithmetical or graphical operation. This is illustrated in the following numerical example.

The double throw, 8-cyl compressor of the preceding example is driven by a hermetic motor. Determine the effect of motor magnetic pull on the polar force diagram in bearing B.

In this case, the rotor is mounted directly on the compressor crankshaft. Due to unavoidable manufacturing tolerances, the air gap between rotor O.D. and stator I.D. is not uniform. This random eccentricity results in a random radial force causing additional bearing forces

$$F_{EA} = -F_E\left(\frac{e}{L}\right)$$

$$F_{EB} = F_E\left(1 + \frac{e}{L}\right)$$

as shown on Fig. 9A. This magnetic force, F_E (Reference 8), is a function of rotor-stator eccentricity for a given motor-design. Taking a representative value $F_E = 500$ lb for a 30 hp hermetic motor, the additional random force on bearing B of Fig. 9A becomes

$$F_{EB} = 500\left(1 + \frac{5.25}{12.75}\right) \cong 706 \text{ lb}$$

The total resultant force on bearing B is obtained by graphical addition of the force vectors F_{EB} and F_B. This can readily be done, without changing the end points of the polar diagram computed in the preceding example, simply by shifting the origin of the (V.H) coordinate system as shown on Fig. 9B. It is evident that this random magnetic pull has a considerable effect on bearing loads and optimum oil feeder grooves location in the main bearings. The compressor designer has several

means at his disposal to minimize the effect of motor magnetic pull:

a) Reduce the rotor overhang, e, for instance, by counterboring the rotor.

b) Design for maximum concentricity, and all other factors being equal, select a motor with a maximum air gap to reduce the relative rotor eccentricity thereby reducing the magnetic pull, F_E.

CONCLUSION

The computation of the forces acting on various components of reciprocating compressors can be performed accurately and economically on a CPC computer. The cost of setting up and utilizing the computer programs developed in this paper has been fully justified. The CPC program allows rapid and economical study of the effects of various parameters on compressor component design, and frees the engineer from time consuming repetitive computations.

ACKNOWLEDGMENT

The authors are indebted to Mr. J. Coyle, of our computation center, for his help and suggestions.

NOMENCLATURE

A_p Piston area $= \dfrac{\pi D_p^2}{4}$, (sq in.)

$a^{(1)}$ Distance between the centerline of the left-hand crankpin to the left-hand main bearing, (in.)—See Figs. 4 and 8 for sign convention.

$a^{(2)}$ Distance between the centerline of the right-hand crankpin to the right-hand main bearing, (in.)—See Figs. 4 and 8 for sign convention.

C Cylinder clearance factor, clearance volume divided by piston displacement volume, dimensionless.

D_p Piston diameter, (in.).

d Distance between connecting rod center of gravity and crankpin centerline, (in.).

F Resultant force transmitted to main bearings, (lb).

F_A Resultant force from crankshaft to main bearing A, (lb).

F_B Resultant force from crankshaft to main bearing B, (lb).

F_c Connecting rod force, (lb).

F_{cp} Resultant force from rod to crankpin, (lb).

F_{cw} Counterweight force per cylinder to offset reciprocating portion of inertia forces, (lb).

F_i Total reciprocating inertia force per cyl, (lb).

F_p Absolute value of a piston assembly reciprocating inertia force, (lb).

F_{RR} Absolute value of the total reciprocating inertia force per cyl, (lb).

F_{RO} Connecting rod rotating inertia force, (lb).

g Gravitational acceleration constant, 32.3 ft/sec.^{2}

K Reciprocating counterweight factor, dimensionless.

k Throw index, k = 1, or 2.

L Distance between two main bearings A and B, (in.).

l Connecting-rod length, center distance of crankpin bore to wristpin bore, (in.).

m Number of cyl per throw.

N Compressor speed, (rpm).

n Gas expansion polytropic exponent.

n^1 Gas compression polytropic exponent.

P_d Compressor discharge gas pressure, (psia).

P_r Compressor cylinder differential gas pressure, (psi).

P_s Compressor suction gas pressure, (psia).

r Crank radius, (in.).

R_A Reaction force from bearing A to crankshaft, (lb).

R_B Reaction force from bearing B to crankshaft, (lb).

S Piston side thrust, (lb).

S^1 Wristpin force, (lb).

T Torque, (lb-in.).

v Cylinder volume, (cu. in.).

W_c Total connecting rod weight per cyl, (lb).

W_{cw} Equivalent reciprocating counterweight per cylinder, (lb).

W_p Piston assembly weight per cyl, (lb).

W_{RR} Total reciprocating weight per cyl, (lb).

W^1_{RR} Reciprocating portion of the connecting rod weight, (lb).

W_{RO} Rotating portion of the connecting rod weight, (lb).

x Cylinder volume factor

$$= 2C + 1 - \cos\alpha + \frac{\lambda}{2}\sin^2\alpha$$

Z Oscillating acceleration factor $= \cos\alpha + \lambda\cos 2\alpha$

α Crank angular position, positive in direction of rotation, (radians).

$\Delta\alpha$ Crank angle increment, (radians).

β Connecting rod angle, positive during the expansion and suction stroke, (radians).

γ_o Angle between the reference (or first) cylinder axis and fixed vertical axis of the machine frame, positive opposite direction of rotation, (radians).

$\Delta\gamma$ Angle between two consecutive cylinders, positive opposite direction of rotation, (radians).

ε Angle between two throws, positive in the direction of rotation, (radians).

λ Crank arm to connecting rod length ratio

$$\lambda = r/l$$

ξ Compressor force distribution ratio on main bearing

$$\xi = 1 - \frac{a}{L}$$

ω Angular velocity of the crank, (radians/sec.).

SUBSCRIPTS

$_X$ X component in rotating (XY) coordinate system.

$_Y$ Y component in rotating (XY) coordinate system.

$_H$ H component in fixed (HV) coordinate system.

$_V$ V component in fixed (HV) coordinate system.

See Fig. 2A and Fig. 7 for sign conventions of (XY), and (HV).

SUPERSCRIPTS

^{E} Electrical pull

$^{(1)}$ first throw

$^{(2)}$ second throw

$^{(k)}$ k throw k = 1, 2.

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Evaporative cooling

for common storages of fruits and vegetables

Fruits and vegetables must be stored in quantity every fall, in order to prevent a market glut at harvest time and to insure a regulated supply throughout the selling period. This paper deals with the improvement of common storage through the use of the simple principle of evaporative cooling. Thus, high humidity to prevent shrinkage and lower temperature to reduce respiration are achieved, with the additional benefit of fresh air to carry away ripening gases and to prevent odor.

ROBERT S. ASH

Storage of fall and winter fruits and vegetables offers a practical method of minimizing price fluctuations in the market place. Ideally, produce should be packed and shipped immediately after being harvested. However, a large portion of fall and winter crops are held in storage to provide a regulated supply to the market throughout the winter and spring.

When fruits and vegetables are harvested they continue to behave like living organisms. Similar to all living things, they carry on the process called respiration. The more rapidly it takes place, the more quickly the produce will deteriorate. Since respiration varies with the temperature, lowering the temperature results in a slower rate of ripening and food materials, flavor, fresh appearance and general quality are conserved.

If fruits and vegetables are put in common or air-cooled storages (insulated buildings cooled during the fall months by introducing night air), the temperature of the product in storage is usually close to the prevailing mean outside temperature[4, 8, 13, 21, 23, 26]. During the months of Sept. and Oct. when there are periods of high temperature the produce ripens prematurely. It also wilts and shrivels from becoming too dry. (A 40-lb bushel of apples can lose as much as 2 to 4 lb, and this loss of weight represents water evaporated from the fruit[23].)

If little fresh air is introduced into the storage, carbon dioxide and ripening gases which evolve from the respiration process will not be carried away. Another problem in the case of apples may be the presence of esters which will cause storage scald. There may also be the problem of odors absorbed from the storage room.[23] Although the picture for apples has been described, the storage problem of other fruits and vegetables is similar.

These common storages for fruits and vegetables can be improved considerably through the use of the simple principle of evaporative cooling, a form of air conditioning based upon the evaporation of water. Evaporative cooling involves what is known as an adiabatic exchange of heat. The sensible heat of the air is reduced proportionally to the amount of evaporation that takes place. The water assumes the wet bulb temperature of the air and cooling proceeds, with the enthalpy or total heat content of the air remaining essentially constant. Humidification occurs as a result of the vapor pressure exerted by the water, which is higher than that corresponding to the entering air dew point. In the process, the dew point temperature rises and the dry bulb temperature falls, with the wet bulb temperature remaining constant. The maximum possible dry bulb temperature reduction is the difference between the entering air dry bulb and the wet bulb temperatures, and this is usually referred to as the wet bulb depression. Were it possible to cool the air an amount equal to the wet bulb depression, the air would be completely saturated. Since this is not the case in actual practice, evaporative cooling systems operate at something less than 100% effectiveness.

In this connection, the temperature of the air resulting from the evaporative cooling process should never be considered as the storage temperature. The latter will be several degrees higher. The temperature difference between the storage room temperature and the evaporative cooler discharge temperature is usually called the diffusion temperature. It depends upon the heat load and the amount of cool air discharged into the storage. In calculating the heat load no attention need be paid to the latent load of respired water by the commodity, as the system operates on the basis of 100% fresh air and any moisture pick up is discharged to the outside. Furthermore, in common storages for fruits

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Robert S. Ash is Assistant to the President, International Metal Products Co. This paper was presented at the ASHRAE annual meeting at Lake Placid, N. Y., June 22-24, 1959.

and vegetables any increase in relative humidity is desirable.[3] Fig. 1 is a skeleton psychrometric chart on which is illustrated a typical evaporative cooling problem involving a heat load that is characteristic of a potato storage house during the early fall, when it is desirable to keep a potato temperature of about 60 F to encourage wound healing.

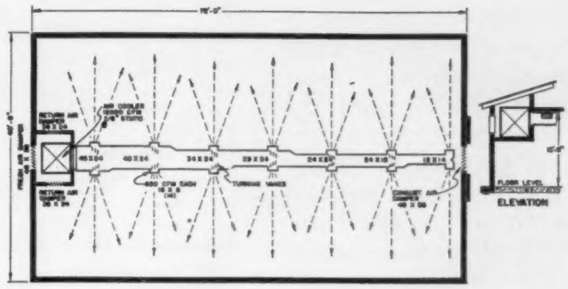
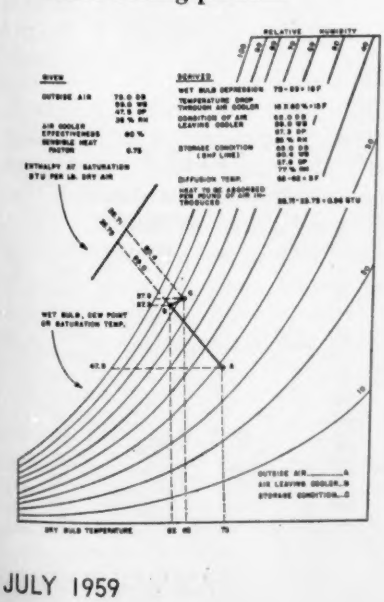


Fig. 2 Floor plan – apple storage, 15,000 bushel capacity

Apples must be harvested at the correct stage of maturity and then be stored immediately for best results.[11,26] When the nights are cool, apples picked in the late afternoon can be left in the orchard over night to lose some of their field heat. To minimize loss of moisture, both the apples and the field crates in which they are stored should be wetted down at time of storage. An air space of 4 in. or more should be left along each wall, and stacking should not be permitted higher than the bottom of the cooling system ducts. The crates should be so stacked that air may circulate between them. A few inches of space left between each tier insures a more rapid air movement through the stacks. Lines painted on the floor of the storage room to indicate the spaces for the rows of boxes are helpful in storing the fruit.

Temperature in the storage should be reduced as rapidly as possible to the ideal apple storage temperature of 31 to 32 F.[13,21,22,33]

Fig. 1 Psychrometric chart illustrating a typical evaporative cooling problem



This can usually be attained by the middle of Nov., as the daily minimum temperature in most storage areas is below this level by that time of the year. High relative humidity is also of vital importance in keeping apples in prime condition. A relative humidity of from at least 85 to 90% should be maintained. This is the approximate water content of apples, and the pull on the fruit for moisture or the evaporative power of the air will be slight and shrinkage of the fruit will be correspondingly lessened.[11] If the temperature is above 36 F such as when the storage is being filled in Sept. and Oct., the relative humidity should be from 90 to 95%. An atmosphere of 95% or higher is necessary to entirely prevent moisture loss. Humidity as high as this, however, favors mold growth on the fruit, crates and walls of the storage.[29]

Evaporative cooling systems for apple storages should be designed to distribute the cool air to all parts of the storage. The evaporative coolers may be floor mounted, or located near the ceiling in a fan room. The system should be designed to discharge the air horizontally at the ceiling level. The maximum practical air throw is about 30 ft. In wide storages where the width exceeds 60 ft, it is usually desirable to install two trunk ducts so spaced that the distance to the wall does not exceed the maximum throw. Canvas hung at strategic positions will force the air to travel through the fruit as it moves to the exhaust opening. The space between the outer stacks and the wall should be blocked at the ends when the storage room is not filled. Stacking crates at the end of the space during loading is enough. If the ends are not blocked, the air tends to move out the ends and around the stacks rather than through them.

The evaporative cooler should be sized to give a three minute air change for quick and even cooling. The total volume of the storage should be considered in calculating the size of the system. The heat load on the evaporative cooling system is sizeable as it consists not only of the heat leakage into the storage, but the field heat of apples as well as the heat of respiration. The heat leakage into the apple storage can be calculated like any other building. Field heat removal must be considered as the apples are placed in storage at outdoor temperature.[25] (If the apples have been left in the sun the temperature may be even higher.) Assuming the apples are placed in storage at outside temperature they must be cooled to the prevailing storage room temperature. The specific heat of apples above freezing is 0.87 Btu. A weight of 50 lb a bushel is assumed for apples. This includes the weight of the field crate which also must be cooled. It then requires 50 x 0.87 or 43.5 Btu to cool a bushel of apples 1 F. To cool 1000 bushels 10 F, 43.5 x 1000 x 10 or 435,000 Btu are required. Since this is for 24 hr, the hourly requirements are 18,125 Btu. In addition to the heat leakage into the storage and the field heat of the apples, it is necessary to take into consideration the heat of respiration.

The approximate evolution of heat through the respiration of apples at different temperatures is shown in Table I.[24,29] These figures represent total heat of respiration and therefore include transpiration. In the process of respiration the water that is evolved is contained within the cellular interstices of the fruit, and is released only as a direct result of transpiration.[5]

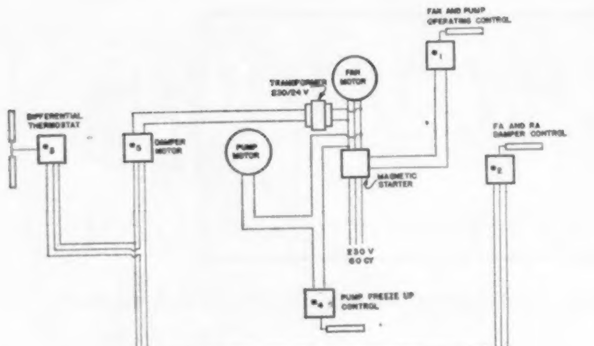


Fig. 3 Evaporative air cooler control system for fruit and vegetable storage

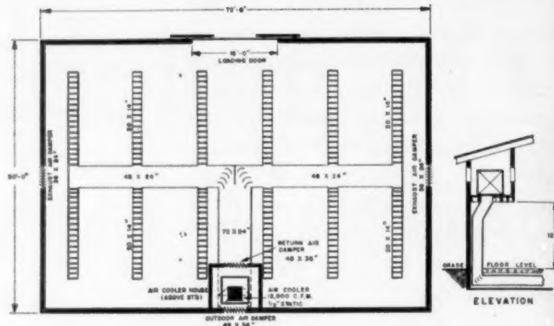


Fig. 4 Floor plan—potato storage house, 25,000 bushel capacity

About 25% of the heat of evolution shown in Table I represents transpiration losses, and can be ignored in calculating the load imposed on the cooling system by the respiration process.[21,22]

Since the degree of cooling feasible is limited by the prevailing wet bulb temperature, the maximum practical size system should be installed to bring down the storage temperatures rapidly and to as close to the wet bulb temperature as possible. In general, a system having a three minute air change capacity is the largest practical system that can be installed. It is customary to provide 2½ cu ft of space for each bushel of apples to be stored. This results in an actual free volume of less than 50%, since the volume of a bushel container excluding the space occupied by the container itself is 1¼ cu ft. An evaporative cooling system that will give a three min. air change actually will result, therefore, in a 1 to 1½ min. movement of air when the storage is loaded. Such a cooling system will improve storage conditions in common storages, so that the length of time in storage can be increased considerably. It is estimated that over eight million bushels of apples annually are held in common storage from one to two months.[7] Common storages with evaporative cooling will give good storage conditions for as long as four months, thus, more than doubling the usefulness of common storages.[9]

Fig. 2 shows the floor plan of a typical 15,000 bushel capacity apple storage with evaporative cooling system. Storage is designed so apples can be stacked to a maximum height of 10 ft. Air handling capacity of system equals

$$\frac{72 \times 40 \times 12\frac{1}{2}}{3}$$

or 12,000 cfm. In sizing the duct system the equal friction method, velocity reduction method or static regain method may be used. The duct system in Fig. 2 was sized by the equal friction method using 0.1 in. friction loss per 100 ft. Table II gives temperature data for the apple storage with evaporative cooling system illustrated in Fig. 2. This plant is located in Northern New Mexico and specializes in dessert quality fruit for gift pack.[9]

Storage temperature averages 6 to 7 F below mean outside temperature which is about the best that can be done with ventilation of common storages.[4] Fruit is kept crisp and firm through Jan. by the evaporative cooling of the storage.[9]

The evaporative cooling system for common storages should be designed so that it is possible: (1) to supply all outside air; (2) recirculate storage air; and (3) circulate a mixture of outside and storage air.[21] The setting of the dampers depends upon the outside and inside temperatures. The setting of the outside air, return air and exhaust dampers may be done manually or by automatic control. Fig. 3 is a wiring diagram for modulating or potentiometer type of control system.

The sequence of operation is as follows:

a. No. 1 return air thermostat set at 32 F starts and stops fan and recirculating water pump of evaporative cooler.

b. No. 4 freeze up thermostat set at 35 F stops pump if recirculated sump water drops to this temperature.

c. No. 2 discharge air thermostat set at 30 F operates No. 5 damper motor to open fresh air damper on a rise in temperature and closes return air damper. Fresh air damper closes on a fall in temperature with return air damper opening.

d. No. 5 damper motor modulates fresh air, and return air dampers at direction of No. 2 proportioning control. Fresh air damper closes and return air damper opens when fan is shut off.

e. No. 3 differential thermostat compares return air temperature with fresh air wet bulb temperature. If fresh air wet bulb temperature exceeds return air temperature, control will close fresh air damper and open return air damper until such time as fresh air wet bulb temperature is below return air temperature. No. 5 damper motor is then returned to control of No. 2 discharge air thermostat.

TABLE I

Evolution of Heat of Apples at Various Temperatures

Temperature F	Heat per ton in 24 hr Btu
32	700
35	1000
40	1500
45	2200
50	3500
55	4600
60	5800
65	6900
70	8000
75	9000
80	10000
85	11000

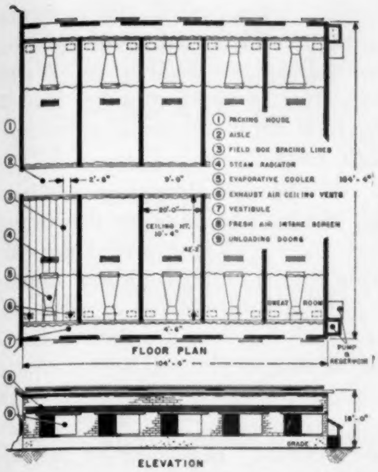


Fig. 5 Coloring and holding building for citrus fruits, 16,000 field box capacity

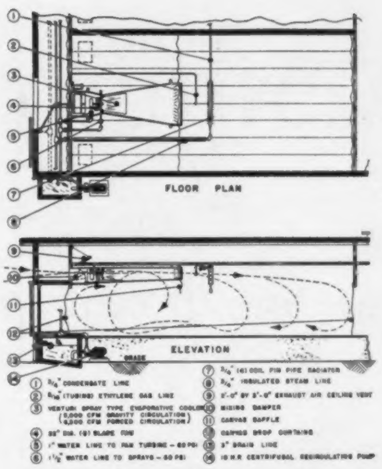


Fig. 6 Sweat room for citrus fruits

f. Automatic shutter or pressure damper opens under static pressure in storage room to exhaust spent air, when system introduces fresh air at time fresh air damper modulates in open position.

In addition to these controls a cycle repeater or time switch may be installed in the system to provide intermittent fan operation when no cooling is required but air circulation is desired. If additional humidity should be needed, spray nozzles can be installed in front of the outlet grilles to open when the fan of the evaporative cooler is in operation. The water lines should be vented so that they will drain free and not freeze up.

Potatoes harvested from late Aug. through Nov. are usually put in storage and may be held for as long as six to eight months.[6,15] The storage period of potatoes is divided into two parts: the wound-healing or curing period and the holding period. The wound-healing period immediately follows harvest. During this period the bruises and other wounds caused during the harvesting operation heal over, and a corky suberin layer forms and reduces the danger of rot. The optimum temperature for this period is 60 F.[10,30] The second stage of storage management is the holding period. In general, the higher the temperature maintained during this storage period, the higher the quality of the tubers. On the other hand, the higher the storage temperature, the shorter the period that the tubers can be kept without sprouting and excessive shrinkage.[28] Table III gives maximum holding periods for various storage temperatures.[17,27]

The proper management of storage means the controlling of temperature, humidity and air circulation so that the stored tubers will retain their texture, flavor and maximum food value with a minimum of loss from rot, shrinkage and sprouting. Potatoes may give off as much as 175 Btu per hundredweight each 24 hrs when stored at high temperatures.[12] Lower storage temperatures result in as little as 20 Btu. Temperature of long term storage should be approximately 38 F.[2,18] A lower temperature causes the starch to turn to sugar, while a higher temperature results in excessive shrinkage through accelerated respiration and transpiration.[2]

Humidity of the air immediately around the potatoes should be as high as possible. In practice, 85 to 90% relative humidity should be maintained.[18] In the fall at harvest time, large volumes of cool air should be drawn through the storage to bring the potatoes to proper storage temperature. After a temperature of 38 to 40 F is reached, cool moist air is needed to prevent the temperature from rising above that point and to minimize shrinkage.[5,24] During the winter when cooling is not needed, proper air circulation is still required to maintain uniform temperature and humidity. This prevents freezing near the walls and sprouting or spoilage by a too high temperature at the bin center.

Evaporative cooling systems for potato storage houses should be designed for through-cooling with the air passing directly through and in contact with the potato pile.[6] Capacity of cooling system should be based on a three minute air change to give quick and even cooling.[3] The total volume of the storage should be considered in calculating the size of the system, so as to provide sufficient cooling if the depth of storage is increased at a future time. Above floor ducts can be used, but under floor ducts are to be preferred as they can also be used for the handling of the potatoes with bin unloaders.[18] The ducts should be 20 in. wide and at least 14 in. deep to allow room for the insertion of the bin unloader conveyors.

Duct tops must be removable and should be 3 x 8-in. lumber to support a loaded truck. Boards should be spaced to provide ½-in. slots. Distributing and delivery ducts act as extended plenums, so it is not necessary to reduce duct depth as the far end of the ducts is approached. Delivery ducts should extend to within 6 ft of the walls to provide uniform air flow through the storage. Since friction loss in the duct system is negligible, the resistance of the potato pile is the principal load on the fan in the air cooler. The resistance is approximately 0.25 in. static pressure (potato storage should not exceed 12 ft depth). The additional resistance of the ducts, inlets, dampers and air cooler pads will add another 0.25 in. or a total of 0.50 in. that the fan must overcome.[24] In large storages it may be advisable to divide the house into two sections to keep the cooling system within practical limits. A single system should not handle over 12,000 cfm of air, if main and distributing ducts are to be of reasonable size.

Fig. 4 shows the floor plan of

TABLE II

Temperature	Sept.	Oct.	Nov.
Mean Outside	61	51	39
Mean Wet Bulb	49	39	29
Average Storage	55	45	32

a potato storage house located on Long Island, New York. The house is designed so potatoes can be piled to a maximum depth of 10 ft. Ducts have been sized by the velocity reduction method using 1000 fpm for trunk duct, 750 fpm for distributing ducts and 500 fpm for delivery ducts. Duct system is laid out so that all delivery ducts are of equal length and spaced on 12 ft centers. The operation of evaporative cooling systems for potato storage houses is similar to that for apple storages. All outside air is normally used during the cooling period. As an example, assume that the desired temperature within the storage is to be 40 F. The system should operate on 100% outside air, if the temperature of the air discharged from the cooler is as cool or cooler than the inside of the storage, and the storage temperature is not lower than 40 F. Should the temperature of the air discharged from the cooler become warmer than the air inside of the storage, the air within the house should be recirculated. If the outside air drops below 40 F, some outside and some storage air should be circulated. During the winter months the air should be recirculated. Continuous recirculation is not necessary, and the air cooler may be put on time switch operation to run intermittently 10 to 15 min. out of every hour.[6]

Citrus fruit – The chief purpose of evaporative cooling, as it is applied to the storage of fruits and vegetables, is to provide an effective yet inexpensive means of improving common storages for fresh produce. It also serves, however, a special and important function in the case of oranges, grapefruit and lemons. It is not uncommon for citrus fruit, although mature and ready for harvest, not to have undergone the natural change in color from green. Color is greatly influenced by temperature variations. And, in the case of Valencia

TABLE III

At 55-60 F potatoes sprout after 70 days
At 50-55 F potatoes sprout after 89 days
At 45-50 F potatoes sprout after 126 days
At 40-45 F potatoes sprout after 200 days
At 35-40 F potatoes keep best
At 30-35 F potatoes turn sweet
Below 30 F potatoes freeze

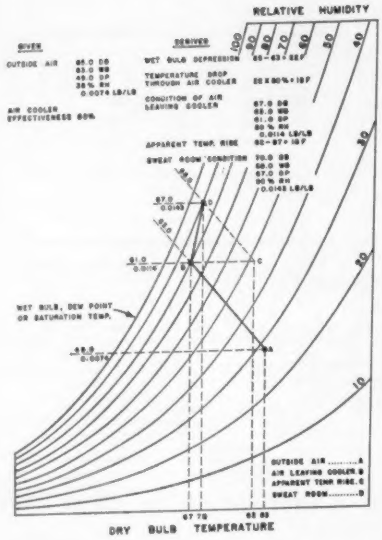


Fig. 7 Psychrometric chart illustrating citrus sweat room conditions

oranges, the green color may even return after the fruit has reached its prime.[19] The consumer, however, expects fruit of characteristic color. This is achieved through a coloring or "sweating" process. Special rooms equipped with evaporative cooling are used to degreen the fruit.[1] Air, with a high relative humidity and at a moderate temperature, is circulated continuously during the sweating operation.

Ethylene gas, the concentration depending upon the variety and the intensity of green pigment in the rind, is discharged into the sweat rooms. The effect of ethylene is mainly to destroy the chlorophyll in the rind and allow the yellow or orange color to become evident. Ethylene accelerates changes that would occur naturally.

Oranges and grapefruit that require degreening are placed in the sweat rooms as soon as delivered to the packing house. Lemons are first washed and separated into color classifications, according to their degree of maturity or color, before being put into the sweat rooms. The fruit should be stacked six field boxes high, with not less than 2 in. between stacks and 3 to 4 in. between rows, so as to allow the air and ethylene to pass freely through the fruit.[19] If the fruit temperature is above 75 F, it should be cooled before applying the ethylene. To avoid excessive shrinkage and other ill effects during the cooling down period, the moisture content in the sweat room must be maintained at a high level. Water should be sprayed onto the fruit if the fruit temperature is extremely high, near 100 F.

Ventilation should be continuous during the cooling down period. In Fig. 5 is depicted the coloring and holding building adjoining a packing house in Phoenix, Arizona. Fig. 6 is a detailed drawing of one of the ten sweat rooms in the coloring building. Each room is approximately 20 x 40 ft with a 10-ft ceiling, designed to accommodate 1600 field boxes each containing 50 lb of fruit. A temperature of 70 F is maintained in the sweat room during the degreening operation with a relative humidity of 88 to 90%. (In the gulf states 80 to 85 F temperature with 90 to 92% relative humidity is recommended.[14,32]) The evaporative cooler is designed to deliver 4 cfm per field box or approximately 6500 cfm at maximum load. 5000 cfm can be obtained with gravity circulation through the induced or venturi action of the spray nozzles in the throat of the cooler.[20]

Additional air capacity is available from the water turbine driven fan if needed. (Fan alone will deliver 3600 cfm.) Each of the six spray nozzles will deliver approximately 4 gph maximum at 60 psi water pressure. High relative humidity is required to minimize shrinkage of the fruit. Although the cooler delivers air at a practically saturated condition, additional free moisture from the spray nozzles is available if necessary to increase the cooling effect, and thereby keep the temperature and humidity in the sweat room within design conditions. The mixing damper in the evaporative cooler, and the two exhaust air ceiling vents which are manually operated, can be manipulated to maintain the desired conditions. Fresh air requirements as low as 300 cfm per 1000 field boxes may prevail when recirculating during degreening and/or winter months.

Water in the recirculating pump reservoir can be heated by means of a steam coil to supplement the fin pipe steam radiator when outside temperatures are low.

(*Continued on page* 85)

Better compressors

will further unite engineering design and user considerations

Any compressor design is inevitably the result of a series of compromises. A unit which offered the highest performance, the lowest operating cost, the lowest cost production, and the longest life, requiring a minimum of service, would undoubtedly be called the best. Before any dimensional decisions, such as bore and stroke, can be established, there must be a determination as to the displacement required to produce the desired refrigeration effect.

The volume of vapor to be displaced by the compressor in cfm may be expressed as:

$$V_e = \frac{200 \text{ x capacity in tons x v}}{(h_s - h_f \text{ x VE})}$$

Where:

h_s = Enthalpy of vapor at temperature and pressure at compressor suction in Btu/lb

h_f = Enthalpy of liquid to expansion valve in Btu/lb

v = Specific volume of vapor at compressor suction in cu ft/lb

VE = Volumetric efficiency in %

In analyzing each value in this formula, we find for any given set of conditions, all values, with the exception of the volumetric efficiency, are fixed by the characteristics of the refrigerant. The volumetric efficiency is the only variable.

In order to be able to design a compressor with sufficient displacement to produce a desired capacity expressed in Btu's or tons, one must have a thorough understanding of the overall compressor volumetric efficiency. Since volumetric efficiency is always a fraction, the actual compressor displacement must be higher than the theoretical displacement.

Overall volumetric efficiency is a combination of factors: Effect of clearance volume and compressor ratio, wire drawing through compressor valves, cylinder heating, blow-by, state of vapor at compressor intake, and compressor speed.

The overall volumetric efficiency cannot be calculated accurately. The final evaluation must rest on actual tests. However, the engineer can direct the design so that the performance can be predicted with a fair amount of accuracy.

Before discussing the individual factors affecting overall volumetric efficiency, let us see what happens during a complete cycle.

1 Compression (Fig. 1A). Assume 40 psi suction and 180 psi discharge. Compression starts at suction dead center and 100% cylinder volume. Cylinder pressure equals suction pressure. Curve A-B represents the pressure increase during the compression stroke. At point "B" cylinder pressure has reached discharge pressure. The cylinder volume has been reduced in this hypothetical case to 20%. The discharge valve is still closed.

2 At this point (Fig. 1B), assuming no pressure drop through

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discharge valve, the valve opens. B-C represents discharge. If there were no clearance space, line B-C would end at 0 cylinder volume.

For mechanical reasons, all reciprocating compressors have a clearance between piston at top dead center and top of cylinder. Discharge ports and other gas pockets add to the clearance volume. That is why at the end of compression stroke there is some vapor left in the remaining cylinder space.

3 (Fig. 1C) The piston has reached compression top center and is moving in opposite direction (suction stroke). The discharge valve is closed. The vapor trapped in the clearance space re-expands, the suction valve will not open until the pressure in the cylinder space has been reduced to the system suction pressure, assuming again, for the time being, no

Focusing on the medium sized, reciprocating compressor, of 10 to 100 ton capacity, this discussion evaluates such equipment from the practical standpoint of the buyer. The latter has such questions as, how much will it cost?; how much will it cost to operate?; how long will it last?; and, what service does it require?; all things which the design engineer needs to keep well in mind.

Frank Raufeisen is Application Engineer in the Air Conditioning Div of the Bell & Gossett Company. This paper was presented as "Practical Evaluation of Design Features in Modern Compressors" at the annual meeting, March 1959, in Los Angeles of ASHRAE Region IX.

pressure drop across suction valve. Curve C-D expresses the line of re-expansion.

The theoretical volumetric efficiency due to clearance can be expressed as the percentage of theoretical piston displacement represented by the actual volume of vapor drawn into the cylinder space.

4 At point "D" the suction valves open admitting fresh vapor into cylinder space. Line D-A completes the suction stroke. It is now clear that the net volume of vapor drawn into cylinder space is only 80%.

5 (Fig. 1D) This curve shows the true cycle. Pressure drops across the suction, and discharge valves distort the theoretical suction and discharge lines. For a fixed clearance, the compressor capacity decreases as the compression ratio increases, until the re-expanded vapor will fill completely the cylinder space. In this case no fresh gas is drawn into the cylinder space any more.

6 (Fig. 2) The theoretical volumetric efficiency varies with the clearance volume. Curve C′-D′ shows the effect of increased clearance volume. Area A-C-C-D expressing compressor capacity and is reduced by area D′-C′-C-D.

Clearance volume alone cannot be made the criterion for good compressor design. Extra valve area quite often will improve overall volumetric efficiency in spite of the fact that the clearance volume is increased by doing so. With today's modern manufacturing facilities and compressor design experience, it is possible to hold the clearance volume to less than 3%.

Cylinder heating—During compression the refrigerant vapor will absorb heat if the cylinder wall temperature exceeds the vapor temperature. Cylinder heating depends on heat dissipation, friction, piston ring pressure and state of cylinder surfaces. Excessive friction can be created by too short a connecting rod. Stroke to connecting rod length plays an important part. For good design, connecting rod L = 3 X stroke.

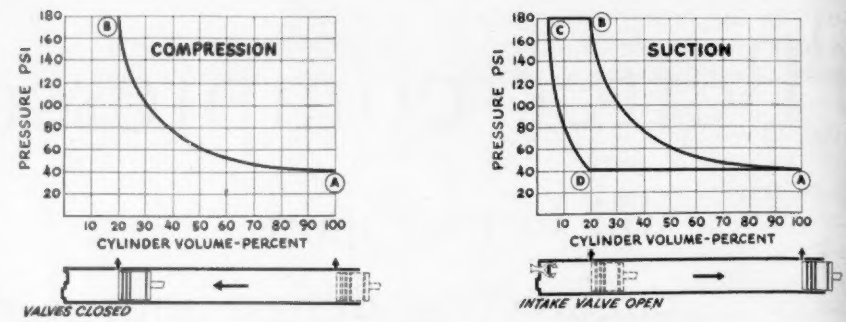
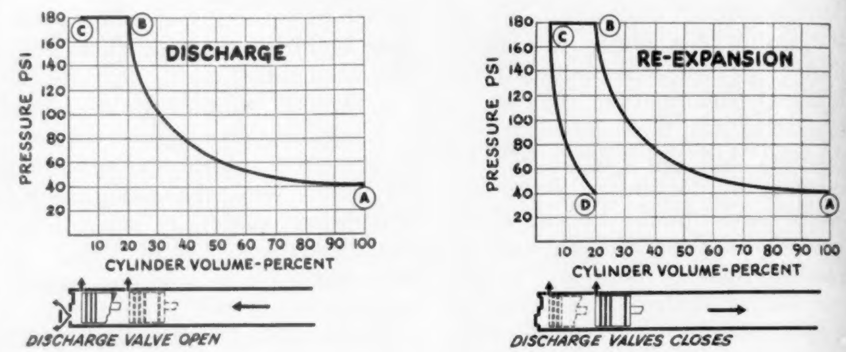


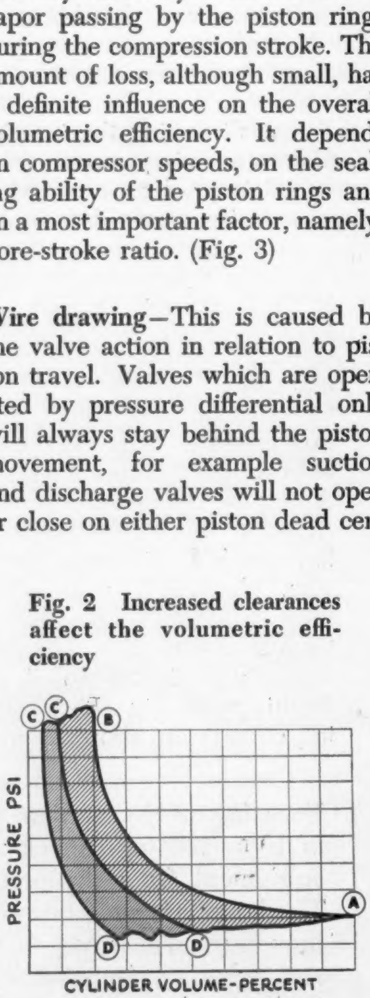
Fig. 1 Four strokes complete the basic reciprocating compressor cycle with variants discussed in the accompanying text



Blow-by—Blow-by is the volume of vapor passing by the piston rings during the compression stroke. The amount of loss, although small, has a definite influence on the overall volumetric efficiency. It depends on compressor speeds, on the sealing ability of the piston rings and on a most important factor, namely, bore-stroke ratio. (Fig. 3)

Wire drawing—This is caused by the valve action in relation to piston travel. Valves which are operated by pressure differential only will always stay behind the piston movement, for example suction and discharge valves will not open or close on either piston dead center. The suction valve will close after the piston has already returned to the discharge stroke, or the discharge valve will close after the piston is already in the suction stroke (Fig. 4)

Fig. 2 Increased clearances affect the volumetric efficiency



State of vapor at compressor inlet—The overall volumetric efficiency is influenced definitely by the state of vapor entering the cylinder space. Volumetric efficiency improves with an increase in vapor superheat. This fact can readily be noted in the original formula for determining the compressor swept volume. While the enthalpy of the vapor increases, the specific volume also increases; however, at a lower percent rate. There are also limitations on the amount of superheat or temperature of the vapor entering the compressor. They are generally set at 65 F vapor entering the compressor, when using Refrigerant-12, and 15 F superheat, regardless of saturated suction temperature, when using Refrigerant-22. These limitations are required to prevent cylinder overheating.

The volumetric efficiency will drop off sharply when the compressor handles wet gas. In this case a certain amount of liquid refrigerant will remain in the clearance space at the end of the com-

pression stroke. It will evaporate during the suction stroke and, therefore, reduce greatly the volumetric efficiency.

Compressor speed – The trend towards higher speeds is dictated by the desire to reduce cost, weight and size. However, with increased speed and, consequently, smaller physical piston sizes, it has become increasingly difficult to find sufficient room for suction and discharge valves. Losses caused by wire drawing will increase with speed, while cylinder heating and blow-by will decrease.

Curve VE - VS - CR – This curve shows the relationship of all the factors mentioned to one and the other, also the theoretical volumetric efficiency and the actual overall volumetric efficiency.

Conclusion–"How does volumetric efficiency help in evaluating compressors?" From it, one can obtain a fairly accurate picture of the machining qualities used in manufacturing the compressor. With modern machine tools and proper valve design, one should maintain a clearance volume of 3%.

Compressor speed – The 1750 rpm compressor speed has become standard practice, though there may be further increase in compressor speed following the search to reduce weight, size and cost. Experience shows that compressors running at 1750 rpm's or higher have as good a life as belt driven, slow speed compressors. The primary factor in limiting speed is the difficulty in finding sufficient space for valve arrangements. This same problem prevails in regard to piston speed.

Piston speed – Theoretically there is no limit to piston speed. For a given piston displacement, the piston area will decrease with increase in piston speed. The limiting factor again is the valve areas which can be provided. It is the ratio between piston area and valve area that limits the piston speed. Well-designed compressors have a maximum piston speed of about 800 fpm. The stroke is a function of compressor speed and piston speed only.

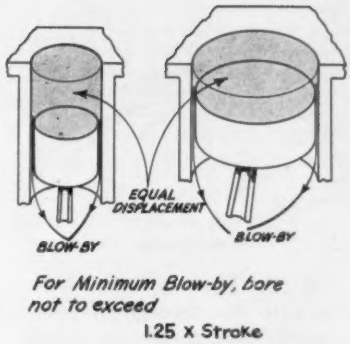


Fig. 3 Minimum blow-by is a direct influence on volumetric efficiency

Valve area – As mentioned, the physical arrangements of the suction and discharge valves become increasingly difficult as the piston diameter decreases. An ideal valve should meet, among others, the following requirements: Largest possible restricted area, of shortest possible length, and be as straight a gas flow as possible, light weight, low lift, yet quick acting, sturdy, inexpensive and tight seating.

The first and second items are the most limiting ones. It is most desirable to have as large a restricted area as possible. Too small an area will result in high gas velocities with resulting loss in power and possible excessive noise. The maximum velocity permissible depends only on how much of a loss the designing engineer is willing to take. Generally, a gas velocity of approximately 6000 fpm will result in no material loss in VE or increase in horsepower.

Over the years, there have been many variations of valve designs with which we are all familiar. No attempt shall be made here to analyze the more ultimate details of these valves.

Valves generally have a lift of approximately 0.060 to 0.075 in.

PISTON DISPLACEMENT

Bore and stroke – The designer of a compressor thus faces the ideal bore and stroke ratio, number of cylinders to match a given capacity, compressor speed and operating conditions, and his work is more or less a trial and error procedure. As discussed, stroke is a function of compressor speed and piston speed. Bore should not exceed 1.25 X stroke. It now becomes a simple matter of arithmetic to determine the physical dimensions of bore and stroke for a given set of dimensions. The mechanics involved may be best illustrated by an example.

Example: Determine: Total compressor volume, stroke, bore and number of cylinders.

Assumption: Refrigerant-12, compressor speed 1750 rpm, piston speed 800 fpm, volumetric efficiency 85%, 40 F saturation suction temperature, 105 F saturated discharge temperature, and compressor capacity – 50 ton.

Starting with our original formula:

$$V = \frac{200 \times \text{Capacity} \times V}{(h_s - h_f) \times VE}$$

For VE, the overall volumetric efficiency, a reasonable figure must be assumed. 85% is a reasonable figure.

There would be a theoretical cfm per ton for 40 F @ 150 F of approximately 3.2

For 50 ton 3.2 x 50 = 160 cfm

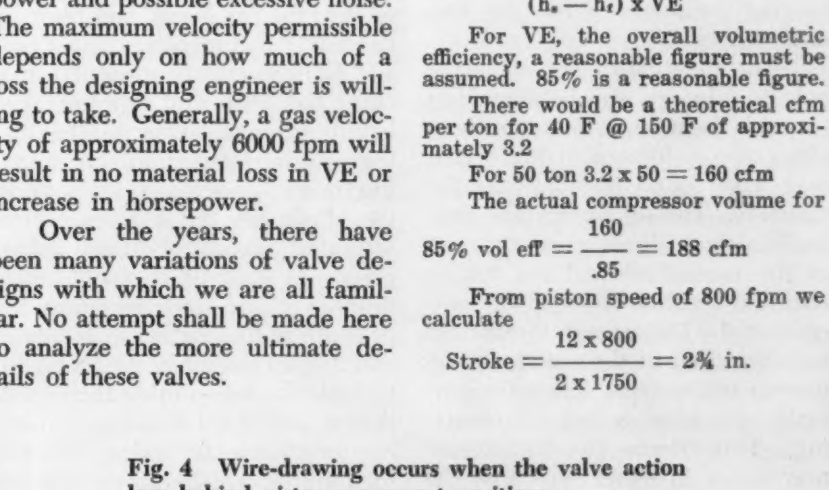
The actual compressor volume for

$$85\% \text{ vol eff} = \frac{160}{.85} = 188 \text{ cfm}$$

From piston speed of 800 fpm we calculate

$$\text{Stroke} = \frac{12 \times 800}{2 \times 1750} = 2\tfrac{3}{4} \text{ in.}$$

Fig. 4 Wire-drawing occurs when the valve action lags behind piston movement positions



For first trial, we assume

Bore = 1.25 x 2.75 = 3.3 in.

We now have a stroke of 2¾ in. and a bore of 3.3 in., or a piston displacement of 23.6 cfm at 1750 rpm.

Number of cylinders will be

$$\frac{188}{23.6} = 7.95 \text{ or } 8 \text{ cylinders}$$

The bore stroke ratio is

$$\frac{3.3}{2.75} = 1.2$$

As the result of weighing all these considerations we need not be surprised to find that several modern compressors, of at least four manufacturers, have either the same bore and stroke or truly close ones.

Lubrication—All refrigeration compressors require lubrication for the bearings, and this generally is accomplished with a pressure lubrication system. Under this system, oil under pressure is forced into all bearing areas. The main bearings are either supplied individually through piping within the compressor directly from the oil pump or the oil is fed through by the rifle drilled holes in the shaft. Quite often the connecting rods are also rifle drilled and thus oil is supplied to the wrist pins. Supplying oil under pressure to all the bearing areas is desirable for several reasons.

The function of the oil is not only to lubricate the bearings, but to cool them and to keep them clean of any foreign matter which may have found its way past the strainers. The oil pumps are generally driven directly off the end of the crankshaft and are totally enclosed within the compressor crankcase. The pumps themselves may be either of the reciprocating, gear or rotary type. They are generally self-priming and self-reversing. This means the compressor may rotate in either direction. On some belt-driven machines this is, however, not the case and careful attention must be paid to the manufacturer's instructions.

Oil return check valve—It is somewhat unfortunate that not all oil remains in the crankcase at all times since the refrigerants are fairly miscible with oil. These facts create problems. During normal operation, a small amount of oil

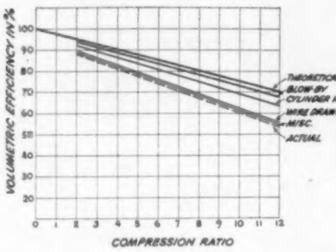


Fig. 5 Various factors attenuate the theoretical volumetric efficiencies to result in actual ones

passes by the piston rings and is discharged into the hot air line and on into the system. If the system is designed correctly, the oil will return to the compressor suction along with the suction gases. Unless provisions are made to allow this oil to return to the crankcase, it will be only a matter of time until all the oil has left the crankcase. The compressor designer, therefore, includes an oil return check valve between the compressor suction and the crankcase. This check valve will allow the passage of oil into the crankcase and at the same time allow the crankcase pressure to be at suction pressure by passing the blow-by gases through the same opening. There is, therefore, a flow of oil into the crankcase, and blow-by gases into the suction.

Certain events occur if a machine has been shut down for any length of time. Due to the fact that oil and refrigerant are miscible, there is no longer only oil in the crankcase, but a more or less saturated mixture of oil and refrigerant. As soon as the compressor turns over and starts to pump, the pressure in the crankcase decreases and the refrigerant in the oil begins to boil off. A condition then exists that is called oil foaming. Unless the crankcase is sealed off, this foaming oil will be drawn into the compressor suction. There is, therefore, an oil check valve. This check valve, however, may not seal tightly; otherwise, the crankcase pressure may climb to the discharge pressure and the oil from the system would never return to the crankcase. The total function of the oil return check valve may be summed up as follows: Allow oil to return to the crankcase during operation, allow the blow-by gases to escape from the crankcase to the suction, seal off the crankcase during start-up, and slowly bleed off crankcase pressure after start-up. This check valve may be only a 25c item, but it is certainly an important one. There are some safety design features in modern compressors that warrant mentioning.

Relief valve — In order to prevent any damage to the compressor in the event a machine should be operated with the discharge line valve closed, an internal relief valve is provided. This relief valve is generally set at 210 psig and will relieve the discharge pressure directly into the compressor suction. Previously, this relief valve has been mounted on each compressor head with the disadvantage that the refrigerant charge will be lost should the relief valve open. Even if the machine is shut off immediately, the relief valve will continue to discharge for a while until it reseats itself.

Shaft seal — On an open type machine, where the shaft protrudes through the crankcase, some means must be provided to prevent the loss of refrigerant to atmosphere. Fig. 6A shows a typical shaft seal used practically on all refrigeration compressors. The actual sealing occurs on two places — around the shaft and at the seal face, between the rotating nose piece and seal face. Some manufacturers keep the entire seal housing under oil pressure to prevent the seal nose from running dry. Also, it is easier to seal oil than gas. Basically, the same arrangement, except in double, is employed in the dual shaft seal shown in Fig. 6B.

Cylinder liners—Virtually all modern compressors employ removable cylinder liners. Incorporated into the liners are the suction and discharge valves. The advantages of service, stocking of parts and interchangeability of parts is seen readily.

Bearings—Sleeve bearings are used almost exclusively in modern compressors. They may be either of the bronze type or steel backed aluminum. Sleeve bearings, in contrast to anti-friction bearings, have proven to have a longer life and to

be quieter. Bearing pressures are generally held below 500 psi of projected area. Bearing adjustments are virtually a thing of the past, due to the trend to precision bearings. Quite often the bearings are pressed into the bearing housings first and then line bored to assure perfect alignment.

Shafts—Shafts have in general been made of forged steel. Today, however, most of the shafts are alloy cast-iron, such as meehanite or gunite. There are several reasons for this. First, they are lower in cost than forgings. Second, they have superior dampening qualities. Third, they have a lower notch sensitivity. These alloy cast-iron shafts have excellent wear resistance in journals, but must be highly polished to prevent scratching of bearings. With modern machining facilities, this does not present a problem.

On a 1750 rpm machine, the balancing of the shaft, both statically and dynamically, is almost a must, more so with the continued trend of eliminating massive foundations and a more critical public. Compressors today can be, and are being, balanced to a point where they do not have any more vibration than a well-made electric motor.

Compressor hp—The theoretical hp required to raise the vapor pressure from the suction pressure to the discharge pressure is the isotropic work of compression. For a condition of 40 F/105 F, this would come out to a figure of approximately .80 to .85 Btu/hr/ton. This does not include any mechanical losses such as losses due to pressure drop across valve plate. In practice, modern compressors require approximately 1.05 to 1.10 Btu/hr/ton and are determined in laboratory tests.

Capacity control—In any refrigeration system there must be a balance between the rate of evaporation in the evaporator and the compressor capacity. If a system were to operate continuously at its design condition, there would be no problem. However, this is seldom, if ever, the case. The load on any system varies usually from a full load condition to a small percentage of full load. For example, in air conditioning, the load during the fall months may only be 25% of the load during July and August. Similarly, in water cooling for the process industries, the inlet water temperature to the cooler may vary with the season and rate of production, yet the desired outlet temperature must be maintained at some fixed temperature.

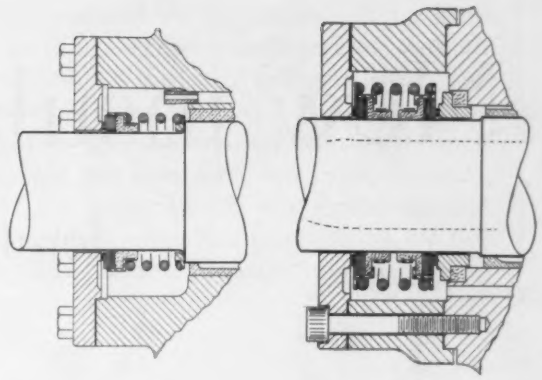


Fig. 6 Shaft seals, customarily of the single type, are used in double forms, too

If the compressor capacity is greater than the rate at which the refrigerant evaporates, the evaporator pressure and temperature are reduced. In the case of a direct expansion air cooler, the air temperature leaving the coil may be reduced to a point where the condensation will freeze on the coil. This ice forming on the coil will reduce the air flow, aggravating the condition by reducing the load and suction temperature even lower, until the unit will shut down on the low pressure cut-out. This will then lead to what is commonly termed "short cycling," a start and stop condition of the condensing unit. This condition is quite unfavorable to the life of the refrigeration equipment and results in high operation cost to the owners. It is evident that some means must be provided to reduce or control the capacity of the compressor. Several means of capacity control have been devised by various manufacturers.

Two speed motors—Since the compressor capacity is proportional to the compressor speed, a two-speed motor driving the compressor may be employed. This would result in either full capacity or reduced capacity, depending on the selected motor speed. This arrangement is satisfactory where the load fluctuation is infrequent or can be anticipated. The two-speed motor may be controlled either manually or with the use of two pressure switches and two-speed magnetic starter. In the latter case, the pressure switches sense the suction pressure. At high suction pressures, indicative of high loads, the machine will operate at high speed. As the suction pressure decreases, the second pressure switch functions to change the motor speed from high to low. An additional low pressure switch will shut the machine down should the load be further decreased.

Hot gas by-pass—One other means to reduce the compressor capacity is to by-pass part of the discharge gas back into the suction of the compressor, which is usually accomplished by the use of a pressure switch and solenoid valve. This arrangement is not entirely satisfactory since the reduction in motor power consumption is relatively small in relation to the capacity reduction. Also, the hot gas returned to the compressor suction causes the compressor to overheat.

Cylinder shut-off — Still another method of capacity reduction is to close off the suction line to one bank of cylinders, so that no gas is pumped by them. This again is accomplished with the use of a pressure switch and solenoid. In employing this system, extreme care must be exercised so that a vacuum is not drawn on the crankcase, resulting in loss of oil.

Cylinder unloading—The most satisfactory method of capacity control on multi-cylinder compressors is that of cylinder unloading. Under this system the suction valves of

(*Continued on page* 87)

New insulation system

for hermetic refrigeration motors

J. P. HURTGEN

A. R. MOUNCE

The electrical insulation system currently used in most hermetic air conditioning and refrigeration motors up to 25 hp in size consists of rag paper ground and phase insulation, Formvar-type wire enamel and braided cotton lead wire insulation. While this system has enjoyed an excellent record of satisfactory performance, it does have disadvantages such as, high initial moisture content, low thermal stability, and evolution of water during operation.

A high initial moisture content compels the compressor manufacturer to perform lengthy dehydration operations which presently require 4 to 15 hr of baking and evacuating at 300 F. The relatively low thermal stability of the Class A (105 C) system, now used, places limitations on dehydration and operation temperatures, as well as limiting resistance to the periodic overload conditions that must frequently be tolerated in air conditioning applications.

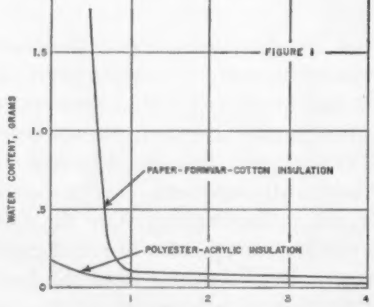
A manifestation of instability, evolved water, promotes overall system corrosion and degradation, and if unchecked can cause freeze-ups if the evaporator temperature is sufficiently low. It is well known that cellulose promotes refrigerant degradation, although the chemical reactions involved have not been elucidated.

It would seem that the value of a different insulation system is highly dependent on the degree to which it lessens the disadvantages, while retaining the advantages, of the paper-enamel-cotton system. To select such a system we devised and carried out a program of screening, investigative and operational tests first on insulation materials, and later on complete insulation systems. Screening tests are rapid and inexpensive measurements of key physical, electrical and chemical properties, and serve to sort out unsuitable materials. Investigative tests probe further into material properties and behavior and often become fairly complex. Motor fabrication trials, effect of heat and atmosphere on electrical properties, and dehydration tests are typical.

Environmental aging and its effect on physical and electrical properties is also studied both with individual materials and combinations thereof. The operational tests involve complete hermetic units and serve to attach practical significance to both questionable and laudable properties and behavior.

At the onset of this evaluation program over 20 basically different wire enamels and sheet materials were studied. Minor changes in material, construction and degree of polymerization swelled this number to over 80. The test results which follow were chosen as being representative of most of the minor variations and in general are those obtained from the modifications which presented the best balance of properties for hermetic applications.

Fig. 1 Stator dehydration comparison for a typical 1 hp rating, 300 F air circulation at 1200 cfm and 78 fpm



MAGNET WIRE

The magnet wire enamels are grouped according to their chemical make-up since certain properties appear to be characteristic of a chemical family and such a grouping makes this more evident. All data shown are for wire having a bare diam of 0.0380 in. and the standard "heavy" build of resin. Resistance to the solvent action of refrigerants was measured by means of Soxhlet extraction, blistering and softening tests which have been described by Wareham.*

J. P. Hurtgen is Development Engineer, Hermetic Motor Dept, A. R. Mounce is Development Engineer, Small Integral Motor Dept, General Electric Co. This paper was presented at the 45th Semiannual Meeting of ASRE in New Orleans, La., December 1-3, 1958 as part of the Domestic Refrigerator Engineering Conference.

* W. W. Wareham, General Electric Co., "Screening Tests for Hermetic Wire" paper presented at First National Conference on the Application of Electrical Insulation, Cleveland —September 3-5, 1958.

Seeking to overcome some of the limitations of the insulations presently used in hermetic refrigerator and air conditioner motors, over 20 basically different sheet materials and wire enamels were evaluated. Describing the screening, investigative and operational tests performed on these materials, and the data obtained from the tests, the authors discuss the properties of the materials that were previously thought to make them unsuitable for this application. They then outline benefits which can be realized by their adoption.

The results are shown in Table I. High extractable contents cause system contamination, a potential failing of magnet wire D, while blistering is at least an indication of unwanted softening of the enamel film, and is felt to be sufficient cause for the rejection of all the polyurethanes and two of the polyesters. Table II lists some physical and electric properties of the remaining magnet wires. Dielectric strength and abrasion resistance were measured in the usual manner. In carrying out the stress-cracking test, a sample of wire is elongated 10% and wound on a mandrel whose diameter varies from one to seven times that of the bare wire being tested (1X to 7X) to simulate winding stresses. The sample is then subjected to 155 C for four hr, cooled and examined visually for radial cracks in the enamel. Magnet wire H, which had been formulated and cured for optimum solvent resistance, looked questionable on this test.

Sealed glass tubes were used to age the remaining wires in a Refrigerant-22 and refrigeration oil atmosphere at three elevated temperatures for several time periods. The typical data in Table III and IV indicate that the acrylic enamel is most stable in this atmosphere, does not evolve visible contaminants and does not greatly accelerate the degradation of either the refrigerant or the oil.

Measurement of solvent resistance of sheet insulation is quite similar to that used for magnet

TABLE I

SOLVENT RESISTANCE OF HERMETIC GRADE MAGNET WIRE

	% Extractables			
	Methanol	Refrigerant-12	Refrigerant-22	Blistering in Refrigerant-22
A—Formvar	5.0	0.64	2.0	Variable*
B—Epoxy	3.2	0.12	.24	None
C—Acrylic I	0.3	0.0	0.0	None
D—Acrylic II	2.0	1.0	1.0	Light
E—Polyurethane I	1.0	0.02	0.03	Heavy
F—Polyurethane II	—	0.1	0.06	Heavy
G—Polyurethane III	—	0.1	0.2	Medium
H—Polyester I	0.0	0.0	0.0	None
I—Polyester II	0.0	0.0	0.2	Light
J—Polyester III	—	—	—	Light

* Depends on formulation and processing

TABLE II

PHYSICAL AND ELECTRICAL PROPERTIES OF HERMETIC GRADE MAGNET WIRE

	Nema Twisted Pair Dielectric Strength (vpm)	GE Repeated Scrape Abrasion Resist. (Strokes)	Extent of Stress-Cracking
A—Formvar	1700	43	None
B—Epoxy	1790	20	None
C—Acrylic I	1750	67	None
H—Polyester I	1720	81	Considerable

TABLE III

AGING AT 150C IN REFRIGERANT-22 AND OIL MAGNET WIRES

	Nema Twisted Pair Dielectric Strength (Kv)			Flexibility After		
	2 wk.	2 mo.	12 mo.	2 wk.	2 mo.	12 mo.
A—Formvar	10.4	7.2	4.9	OK	7X	7X
B—Epoxy	11.3	8.0	6.9	OK	1X	5X
C—Acrylic I	11.2	8.6	5.7	OK	OK	7X

TABLE IV

AGING AT 150C IN REFRIGERANT-22 AND OIL CONDITION OF OIL

	Color of Oil	Contamination
A—Formvar	Dark brown	Brown deposits
B—Epoxy	Lemon yellow	Cloudy oil—no deposits
C—Acrylic I	Light yellow	Clear oil—no deposits

TABLE V

REFRIGERANT RESISTANCE OF SHEET INSULATION

Material	% Extracted by Refrigerant-22	Physical Condition After Refrigerant-22 Blister Test
A—Rag paper	0.30	OK
B—Polyester film	0.0	OK
C—Acrylic resin coated fabrics & mats	0.2	OK
D—Polytetrafluoroethylene coated glass fabrics	0.0	OK
E—Polyethylene film	0.0	OK
F—Irradiated polyethylene film	0.0	OK
G—Polycarbonate film	1.57	OK
H—Epoxide coated glass fabric	2.61	Small blisters
I—Polyurethane coated glass fabrics	0.90	Swelled, small blisters

TABLE VI

OIL RESISTANCE OF SHEET INSULATION, 21 DAYS AT 150C

Material	Condition of Sheet	Condition of Oil
A—Rag paper	OK	Slight yellowing
B—Polyester film	OK	Cloudy
C—Acrylic resin coated fabrics & mats	OK	OK
D—Polytetrafluoroethylene coated glass fabrics	OK	OK
E—Polyethylene film*	Very soft	Slightly cloudy
F—Irradiated polyethylene film*	Very soft	Slightly cloudy

* Tested at 100 C

wire. The results of such tests (Table V) show that materials G, H and I are not suitable. Behavior during immersion in 135 C refrigeration oil for several weeks is significant since the insulation and oil are in intimate contact throughout the life of the unit. As shown in Table VI the polyethylenes are definitely not suitable and the polyester film exhibits partial solubility in oil, a situation that requires further study. Some physical and electrical properties of the remaining sheet materials are shown in Table VII. As expected, the moderate tensile strength coupled with low elongation of the coated fabrics gave rise to considerable motor fabrication difficulties, and only the polyester film proved consistently capable of adaptation to high production techniques. This ability is essential for the motor sizes concerned.

Since the polyester film, material B in the tables, was most promising, its behavior in oil was studied. It was learned that the small finite portion extracted by the oil is stable, insoluble in refrigerants and usually precipitates from the oil in a granular condition. Extraction does not affect film properties since the extractable material is simply a low molecular weight polymer comprising about 1% of the original film weight. Nothing short of tests in compressors is capable of evaluating practically the tolerance of a refrigeration system to this material. Such tests were also needed to determine the rate at which hydrolysis of the polyester film occurs since earlier work had demonstrated that sealed

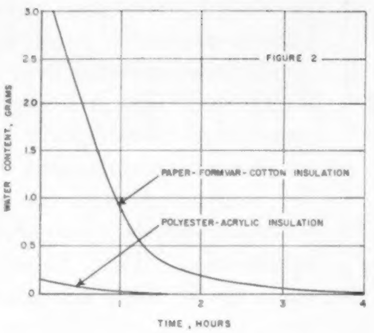


Fig. 2 Stator dehydration, as in Fig. 1, but at 30 μ vacuum

system aging at elevated temperatures rapidly embrittled this film if water was present. The rate and extent of polyester hydrolysis is dependent on the partial pressure of water, the temperature, and the total quantity of water available.** Since these conditions are dynamic in refrigeration systems, the operating unit tests were chosen in favor of continued sealed tube testing.

OPERATIONAL TESTS

Stators insulated with polyester film were installed in four different types of units selected for their high sensitivity to the presence of a solid contaminant, and were operated at average winding temperatures of 266 and 284 F for six months. Both Refrigerants-12 and 22 were used, and the initial amount of circulating water was 30-40 ppm. During this time the power input and starting voltages for all 24 units remained essentially constant, and on disassembly no foreign material was found on any of the bearing surfaces.

Although the oil had removed 1% of the polymer, only minute amounts were found in the capillary tubes and the appearance of the compressor valves was similar to that experienced with paper-enamel-cotton units. None of the polyester film in any of the units became brittle or detectably less flexible. Based on these results we have concluded that neither extraction of the polyester film by oil or its hydrolysis is objectionable when the film is used in compressors similar to the ones we have tested.

The selection of braided sleeving, lead wire insulation and tying cord and thread made from polyester fiber was carried out in a manner entirely analogous to that described above.

SYSTEM PERFORMANCE

Having quantified the superiority of acrylin resin magnet wire, polyester film and polyester fiber, tests on the complete insulation system were in order. After suitable changes in manufacturing techniques and processes, quantities of motors incorporating the new materials were built.

The results of our dehydration tests on complete stators are shown in Figs. 1 and 2. Conventional units must be dehydrated for one to two hr before their moisture content falls below the initial moisture content of units employing the new insulation system. In general, drying the new system to a certain moisture level required only one-third the time needed to reduce the moisture in the paper-enamel-cotton units to the same level.*

Additional tests in operating hermetic units similar to those described above confirmed simpler tests rating the thermal stability of the new system. Measurements of refrigerant oil and insulation degradation indicate the polyester-acrylic system is more stable than the paper-enamel system even when operated at temperature levels 35 F higher than are commonly maintained with the paper-enamel system. One such measurement is moisture evolution during opera-

(*Continued on page* 94)

** J. P. Harrington and R. J. Ward, "Polyester Film Insulation for Hermetic Motors", ASHRAE JOURNAL, April, 1959.

TABLE VII
PHYSICAL & ELECTRICAL PROPERTIES OF SHEET INSULATION

Property	A Rag Paper MD & CMD	B Polyester Film	C Acrylic Coated Fabrics & Mats	D Polytetrafluoroethylene Coated Fabrics
Tensile strength, lb.	190 x 67	17,000	200	200
Elongation, %	5 x 10	100	0	0
Tear strength, grams	720 x 1100	18	900	900
Dielectric strength, vpm	300	2800	500	500
Dielectric constant	—	3.16	4.05	3.0
Dissipation Factor at				
25 C, 60 cps	—	.0021	.048	0.0007
150 C, 60 cps	—	.0064	.75	—
Thickness, mils	15	7.5	10	10

Experimental

Study of grille noise characteristics

B. H. MARVET

Member ASHRAE

To predict characteristic patterns inherent in grille noise, three test grilles shown in Fig. 1 were compared under identical experimental conditions by relating octave-band spectra, loudness spectra in sones, and total loudness in sones, all as functions of the grille face velocities.

The octave-band analysis serves the purpose of indicating the influence of certain variables on the frequency distribution. Loudness, an important criteria of grille noise, provides a standard whereby the test grilles can be placed in some rank and order with respect to an observer.

The chart shown in Fig. 2, devised by the Armour Research Foundation, provided a convenient tool for studying the loudness characteristics of grille noise. By plotting the octave-band data on this chart, the loudness spectra in sones were obtained. The method presented by this chart is based on the psychoacoustic experiments conducted by S. S. Stevens of Harvard University,[1] and is now widely accepted over the older equivalent tone method.

Test equipment and instrumentation – Fig. 3 shows the air supply system used to obtain the required grille noise data. This system met the basic need for a controlled air supply having a low noise level, and an enclosed space of low ambient noise level where a wide range of grille noise intensity could be measured.

The air supply fan was a centrifugal type, single inlet, single outlet, having a 9-in. wheel. It was powered by direct-drive dc motor, whose speed could be varied by means of a rheostat on an adjacent control board. A glass fiber filter was mounted on the inlet side of the fan, with a sheet metal transition, to prevent dirt from being deposited inside the system.

The air measurement duct contained an air-straightener section, Pitot-tube, and thermometer connections, for determining the air flow rates to the test grilles. A hook gauge was connected to the Pitot-tube, to obtain the velocity pressures, particularly at low flow rates.

A sound trap was required to reduce the noise intensity generated by the supply fan before it reached the test grilles. This trap, constructed of plywood, consisted of a series of acoustically-lined 180 deg turns. The first three sections of the trap had two air flow paths, and the last section, four air flow paths. The acoustic insulation used was 1-in. flexible duct liner. All of the exposed insulation surfaces to the air flow were painted to prevent fiber corrosion, and reduce the surface friction factor.

The anechoic chamber, which measured 6 ft 6 in. wide x 7 ft 6 in. deep x 7 ft 0 in. high overall, provided the space for measuring the grille noise intensities. The typical wall construction consisted of glass fiber wedges on the inside, 4-in. thick insulation on the outside, with a 4-in. air space between. The wedges, 8 x 8 in. at the base and 10 in. high, were placed in an alterating pattern. The sound transmission loss through the wall was quite satisfactory for eliminating outside noise. Preliminary tests made when this chamber was first constructed indicated good free-field characteristics for frequencies above 300 cps.

One wall of the chamber was hinged to provide access to the inside. A sheet metal sleeve for holding the test grilles extended through another wall. On the opposite wall a number of wedges were removed to allow the air to exit through an exhaust duct to the outside of the building. This duct, which was constructed of plywood, consisted of two airflow paths lined with 1-in. flexible duct liner. A static pressure tap was added to

Presenting a graphic comparison of the patterns of octave-band and loudness spectra, the author shows what effect fin design, grille face velocity, and static pressure loss have upon the frequency distribution and subjective loudness of grille noise. Three sample grilles of the return-air type, showing sufficient variation in fin shape and spacing to make the study practical, were used in this investigation. The effects of such variables as aspect ratio and mass air flow rates were not analyzed.

B. H. Marvet is a Mechanical Engineer, Div of Design, Tennessee Valley Authority. This paper is based on the author's M.Sc. thesis written for the University of Tennessee, and was presented at the ASHRAE annual meeting in Lake Placid, N. Y., June 22-24, 1959.

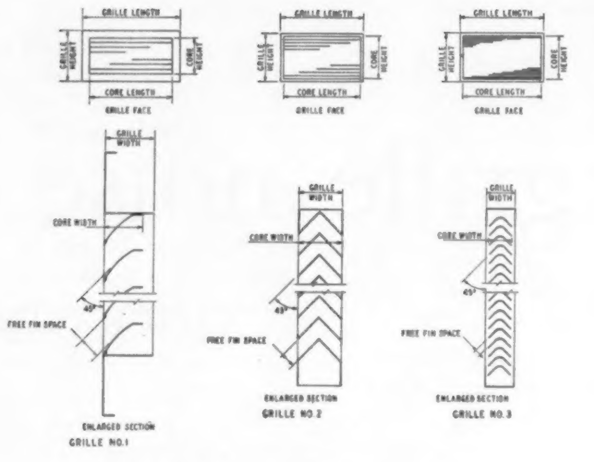


Fig. 1 Grilles used for experiment (See Table I for physical data)

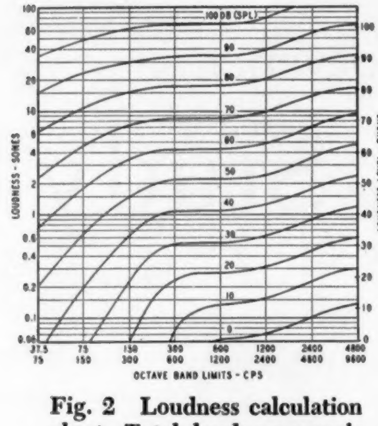


Fig. 2 Loudness calculation chart. Total loudness equals number of sones in loudest band plus 0.3 the sum of sones in other bands

the grille mounting sleeve, for measuring the static pressure loss through the test grilles.

The instrumentation for the noise measurements consisted of a microphone, sound-level meter, and an octave-band analyzer. The microphone was located at the center of the test grille, and 12 in. from it. Although this location avoided the direct discharge of air against it, the microphone was shielded with a suitable wind screen made of a wire frame covered with muslin.

Test procedures – The basic information needed to correlate experimental data involved establishing for each grille the air flow rate, static pressure loss, total noise with grille in place, and background noise.

The air flow rates to each test grille were determined as a function of fan speed, using an 8-point Pitot-tube traverse. A sufficient number of fan speeds were used to provide the calibration curves of air flow rate as a function of fan speed.

The first series of noise measurements were taken with the grille in place, representing the total noise. The octave-band sound-pressure levels, and grille static pressure losses were determined for eight fan speeds ranging from 800-1500 rpm. In addition, the velocity pressure reading at the midpoint of the air flow measurement duct was obtained for each test speed by means of the Pitot-tube and an inclined draft gauge. This pressure was used as a reference point to represent the air flow rate at the particular test speed.

The background noise at the microphone location was simulated by removing the test grille and supplying the same air flow rate, at the same test speeds, as were used with the grille in place. In this manner fan and air turbulence noise could be reproduced at the microphone.

In order to provide the same flow rate for a given test speed, it was necessary to throttle the air flow system until the representative velocity pressure reading at the midpoint of the air flow measurement duct was indicated. The necessary throttling was accomplished by adding a well-insulated enclosure around the filter housing at the fan inlet. The enclosure contained an opening fitted with a sliding panel which served as the throttling device, and reduced effectively the throttling noise before it entered the system.

Generation of noise at the microphone wind screen was considered to be an important source of background noise at the lower frequencies. To produce the same air flow patterns with the grille removed, a nylon cloth deflector was attached to the outlet of the grille mounting duct. Using the same fan speeds and related velocity pressure readings, the background noise octave-band sound-pressure levels were determined at the test location.

Experimental data for the octave-band pressure levels of the grille noise were corrected to obtain the sound pressure levels per sq ft of core area.

The total sound-pressure levels, SPL_t were first corrected for the existing background sound pressure levels, SPL_b. The method that is usually correct is to subtract SPL_b from SPL_t on an energy basis. The number of decibels representing the sound-pressure levels can be converted to relative powers, since sound-pressure is usually proportional to the square root of sound power. If W/W_o is the power ratio for a given number of decibels, and P/P_o is the corresponding pressure ratio, then

$$DB = 10 \log_{10} \frac{W}{W_o} = 10 \log_{10} \frac{P^2}{P_o^2}$$

If the value of P_o, the reference pressure, is 0.0002 microbars, then the above equation defines the decibel unit to indicate sound-pressure level. Thus, SPL_c, the difference between total and background sound-pressure levels of grille noise is determined by taking the difference of the respective power ratios,

$$\text{or } SPL_c = 10 \log_{10} \left[\text{antilog} \frac{W_t}{W_o} - \text{antilog} \frac{W_b}{W_o}\right], \text{db}$$

$$\text{or } SPL_c = 10 \log_{10} \left[\text{antilog} \frac{P_t^2}{P_o^2} - \text{antilog} \frac{P_b^2}{P_o^2}\right], \text{db}$$

$$\text{and } SPL_c = 10 \log_{10} \left[\text{antilog} \frac{SPL_t}{10} - \text{antilog} \frac{SPL_b}{10}\right], \text{db}$$

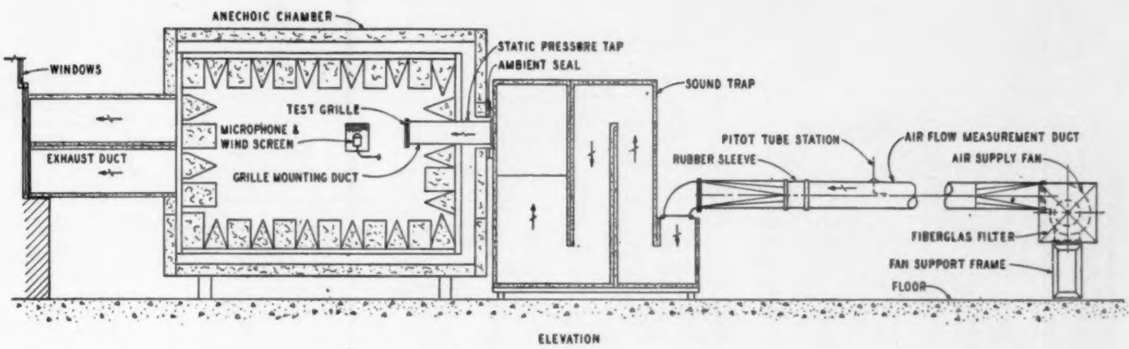


Fig. 3 Air supply system

For example, if the measured SPL_t for a particular octave-band is 85 db, and the existing SPL_b in that band is 80 db, then SPL_c, the noise attributed to the grille is

$$SPL_c = 10 \log_{10}\left[\text{antilog}\frac{85}{10} - \text{antilog}\frac{80}{10}\right] = 83.3 \text{ db}$$

Background noise correction was required in the octave-bands below 300 cps. The higher background noise in the lower octave-bands can be attributed to the predominance of low-frequency noise generated at the microphone wind screen, and to the poor efficiency of the sound trap in attenuating low-frequency fan noise. The correction at the higher octave-bands was essentially equal to zero.

In order to compare the noise levels of each grille on an equal basis, it was necessary to correct SPL_c to obtain SPL_s, grille noise per sq ft of core area. Similar to the background noise correction, the area correction was determined on an energy basis. Thus, SPL_s can be represented by acoustic power per sq ft, as follows:

$$\text{Antilog}\frac{SPL_s}{10} = \frac{\dfrac{W_c}{W_o}}{\dfrac{A_g}{A_o}}$$

where

A_o = one ft^2

A_g = grille core area, ft^2;

therefore

$$SPL_s = 10 \log_{10}\frac{\dfrac{W_c}{W_o}}{\dfrac{A_c}{A_o}}, \text{ db}$$

$$SPL_s = 10 \log_{10}\frac{W_c}{W_o} + 10 \log_{10}\frac{A_o}{A_g}, \text{ db}$$

$$SPL_s = 20 \log_{10}\frac{P_c}{P_o} + 10 \log_{10}\frac{A_o}{A_g}, \text{ db}$$

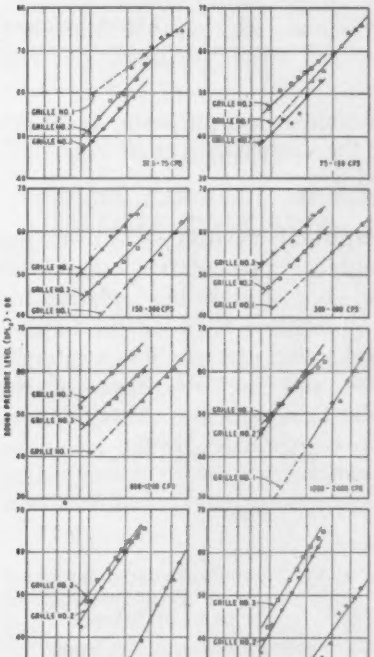
$$SPL_s = SPL_c + 10 \log_{10}\frac{A_o}{A_g}, \text{ db}$$

The decibel correction for area to be added to SPL_c is $10 \log_{10} \frac{A_o}{A_g}$. For example, a 0.5 sq ft grille core, with a SPL_c of 83.3 db for a specific octave-band, would have an area correction of $10 \log_{10} \frac{1}{0.5}$ or 3 db. The resulting SPL_s for the core would be 86.6 db. The same result is obtained by adding the SPL_c for two 0.5 sq ft grille cores. The area correction values for each test grille are tabulated in Table II.

The resultant grille noise data, SPL_s, are presented graphically as a function of grille face velocity for each octave-band as shown in Fig. 4. The curves for Grille No. 1 are extrapolated to the lower velocities, as shown by the dashed portion, in order to allow a wider area of comparison. This extension seems reasonable when considering the general straight-line trend of the curves.

The grille face velocities were determined from the air flow rates and the test grilles free core area tabulated in Table I. The free core areas were measured on a plane perpendicular to the air flow discharge at the grille face, which was taken as 45 F for all of the grilles.

Fig. 4 Comparison of sound-pressure levels of grille noise for each octave band



The octave-band spectra for each grille, at various grille face velocities, shown in Fig. 5, were plotted from the smoothed curves of Fig. 4. By combining the octave-band spectra with the Loudness Calculation Chart, the loudness spectra for each grille were obtained, as shown in Fig. 7. The total loudness curves for each grille shown in Fig. 8 were evaluated from their respective loudness spectra and the summation procedure shown in Fig. 2.

Again, the graphical presentations for Grille No. 1 were extended to the lower grille face velocities

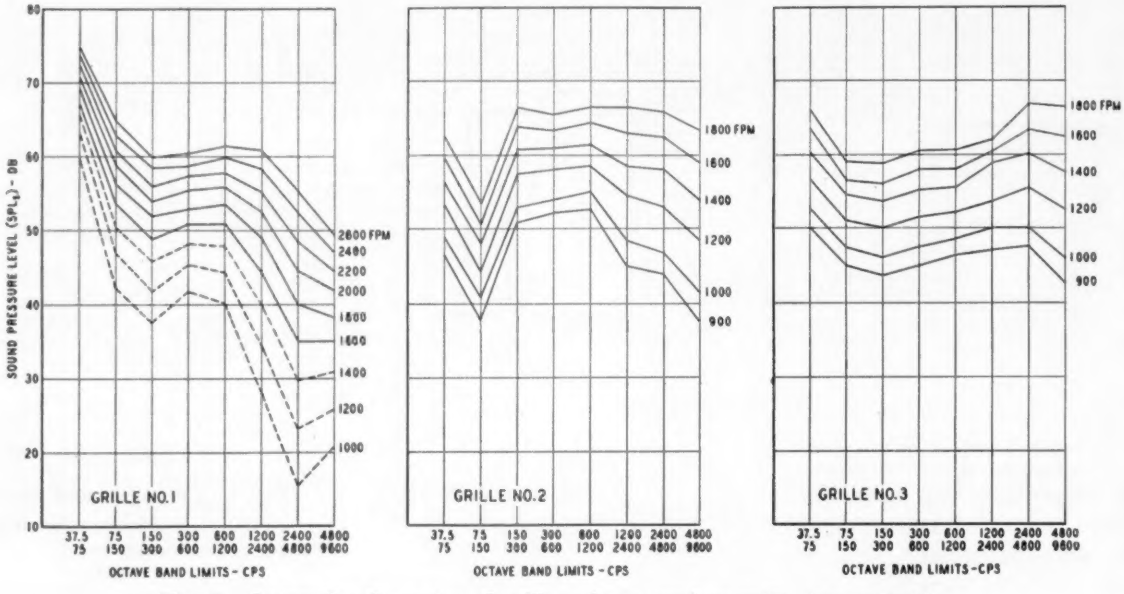


Fig. 5 Octave band spectra of grille noise at various grille face velocities

for comparison purposes, as shown by the dashed curves in Figs. 5, 7, and 8.

Graphical results – From the graphical presentation shown in Fig. 4 a straight-line relationship is established between the octave-band sound pressure levels of grille noise and the logarithms of the grille face velocities. This relationship is expressed by the following equation:

$$SPL_o = a \log_{10} V_f + b, \text{ db}$$

where

a and b = constants dependent upon the grille and octave-band of interest

V_f = grille face velocity, fpm

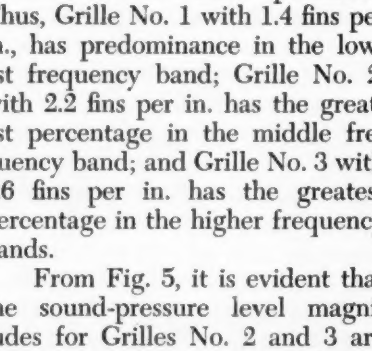
Deviations from the straight-line appear mainly in the octave-bands below 300 cps. Apparently, these deviations can be attributed to the difficulties of measuring the background noise in the lower octave-bands.

Fig. 4 also shows that the slope of the curves increases in the higher octave-band. This would indicate that at the higher frequencies, grille noise increases at a more rapid rate with increasing velocities.

In Fig. 5 which shows the octave-band spectra for each grille, it is observed that as the grille face velocity increases, the spectra contain greater percentages of the higher frequencies. This results in greater predominance of the higher octave-bands in the case of Grilles No. 2 and 3. A comparison of the frequency distribution for the grilles indicates that some correlation exists between the number of fins in the grille core and the predominant octave-band produced. Thus, Grille No. 1 with 1.4 fins per in., has predominance in the lowest frequency band; Grille No. 2, with 2.2 fins per in. has the greatest percentage in the middle frequency band; and Grille No. 3 with 5.6 fins per in. has the greatest percentage in the higher frequency bands.

From Fig. 5, it is evident that the sound-pressure level magnitudes for Grilles No. 2 and 3 are greater, as compared to Grille No. 1, mainly in the higher-frequency bands. The greater acoustic energy contained by the noise of Grilles No. 2 and 3 is an indication of their greater aerodynamic losses due to air flow, which can be represented by the static pressure loss curves shown in Fig. 6.

Fig. 6 Comparison of grille static pressure losses



The differences in pressure losses for the three test grilles reflect the differences in fin design. The fin spacing, shape, depth, and thickness are important factors which affect, to varying degrees, the static pressure losses. It is apparent that more fins per in. result in higher static pressure losses due to more friction surfaces. This fact is substantiated by the higher pressure losses for Grilles No. 2 and 3.

Although Grille No. 2 has less fin per in. than Grille No. 3, its pressure losses are higher. The reasons for this are seen in Fig. 1. The fin shape for Grille No. 2 is a sharp 90 deg. bend as compared with the rounded fin shape for Grille No. 3. In addition the linear distance of air flow through the fins for Grille No. 2 is longer as compared to Grille No. 3.

Another observation is that higher percentages of free core area do not necessarily indicate lower pressure losses. A good example of this is to compare Grille No. 1 with 2 and 3. Although Grille No. 1 has the smallest percentage of free core area, its static pressure losses are much lower. The reasons for this can be attributed to Grille No. 1's smaller number of fins per in., and the smooth 45 deg. bend of the fins.

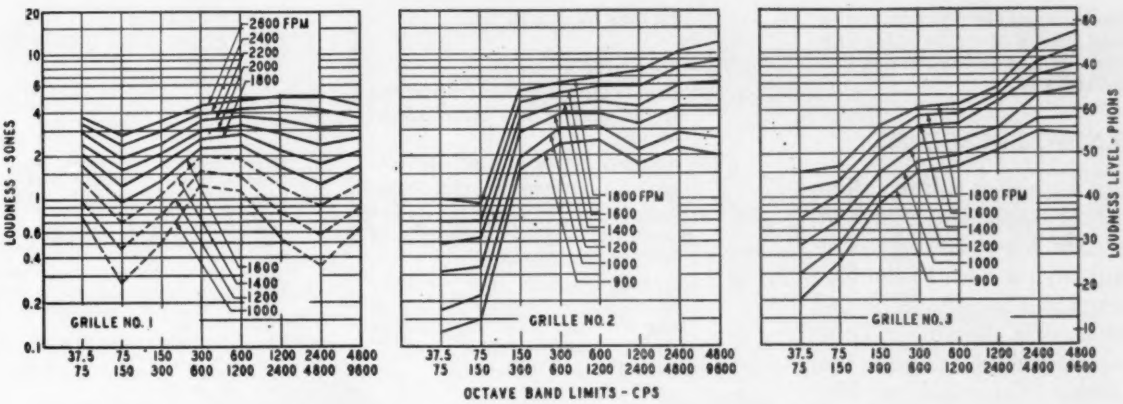


Fig. 7 Loudness spectra of grille noise at various grille face velocities

In Fig. 7 the loudness spectra of noise for each grille are shown. It is apparent that the sound-pressure level magnitudes in the higher-frequency bands are most important. In addition, as the grille face velocity increases, the higher-frequency bands become louder. This is due to the greater percentage of the higher frequencies occurring at the higher grille face velocities.

TABLE I
PHYSICAL CHARACTERISTICS OF TEST GRILLES

Grille No.	1	2	3
Grille length, in.	14	14	14
Grille height, in.	8-3/8	8	8
Grille width, in.	1-1/16	15/16	5/8
Core length, in.	11-1/2	12-15/16	13-1/16
Core height, in.	5-27/32	6-15/16	7-1/8
Core width, in.	7/8	15/16	1/2
Grille area, ft.^{2}	0.813	0.778	0.774
Core area, ft.^{2}	0.467	0.623	0.646
Free core area, ft.^{2}	0.256	0.399	0.401
Free core area / Core area	0.547	0.640	0.620
No. of fins	8	15	40
Fins per in.	1.4	2.2	5.6
Fin thickness, in.	.030	.030	.015
No. of free fin spaces	7	16	41
Free fin space, in.	0.457	0.278	0.108

The total loudness curves for the grille noise presented in Fig. 8 summarizes the loudness spectra. The calculation method used for total loudness indicates that these curves mainly represent the loudest octave-bands. It is important to observe that the strong low-frequency octave-bands that appeared for Grilles No. 1 and 3 had but little effect on the total loudness of the grille noise. A comparison of total loudness for each grille indicates that Grille No. 1 is the quietest, and Grilles No. 2 and 3 are the loudest. At 900 fpm velocity, Grilles No. 2 and 3 are 2.7 times as loud as Grille No. 1; at 1800 fpm velocity, they are 3.2 times as loud.

The total loudness curves of grille noise indicate a rapid rise of the loudness with increasing grille face velocities. A 100% increase in velocity from 900 to 1800 fpm results in a 275% increase in total loudness for Grille No. 1, and a 340% increase for Grilles No. 2 and 3. The more rapid rise in total loudness for Grilles No. 2 and 3 can be attributed to the higher-frequency predominance in their noise spectra.

Observe that the equal values of total loudness for Grilles No. 2 and 3 point out the cumulative influences of static pressure loss and number of fins per in. For Grille No. 2, the higher pressure losses overcome the advantage of its small number of fins; for Grille No. 3 the higher number of fins overcome the advantage of smaller pressure losses. Grille No. 1 indicates that the best fin characteristics for producing a low loudness rating are small pressure losses in combination with a small number of fins per in.

Fig. 8 Comparison of total loudness of grille noise

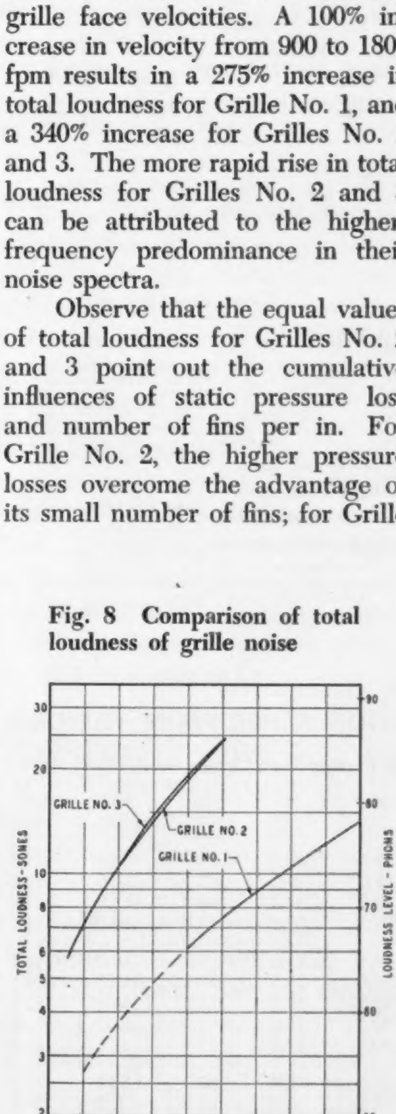


TABLE NO. II
DECIBEL CORRECTION FOR GRILLE CORE AREA

Grille No.	A_c (ft^2)	$10 \log_{10} \frac{A_o}{A_c}$ (db)
1	0.467	3.30
2	0.623	2.04
3	0.646	1.90

In general:

1. Grille face velocity, fin spacing, and static pressure loss are interrelated factors influencing the frequency characteristics and loudness rating of grille noise.
2. Octave-band sound-pressure level of grille noise is a function of the grille face velocity which can be expressed as follows:
   $SPL_o = a \log_{10} V_f + b$, db
3. Increasing the grille face velocity results in a greater percentage of higher frequencies in the grille noise spectrum.
4. Grilles with a larger number of fins per in. tend to have higher-frequency predominance in the grille noise spectrum
5. Grilles with higher pressure losses have greater magnitudes of sound-pressure levels in the high-frequency range of the grille noise spectrum.
6. Loudness rating of grille noise is dependent upon the high-frequency range of the grille noise spectrum.
7. Increasing the grille face velocity results in a rapidly increasing loudness rating of the grille noise.
8. Grilles with a large number of fins per in. or a large static pressure loss, will have a high loudness rating of grille noise.

9. Best combination for producing a low loudness rating of grille noise is a small number of fins per in. and a low static pressure loss.

ACKNOWLEDGMENTS

The author expresses appreciation to Dr. P. F. Pasqua, University of Tennessee, for his encouragement and helpful suggestions; to the following companies for providing the test grilles used in this study: Titus Mfg. Co.; Arolite Co.; and the Barber-Colman Co.; and to Mr. Franklin G. Tyzzer, Armour Research Foundation for use of the Loudness Calculation chart.

NOMENCLATURE

A = core area in sq ft
a, b = constants dependent upon the grille and octave-band of interest
db = abbrev. representing decibels
P = sound-pressure in microbars
SPL = octave-band sound-pressure level in decibels relative to 0.0002 microbars
V_f = average grille face velocity in a plane perpendicular to the air flow discharge at the grille face, in fpm
W = sound-power in watts

Subscripts
$_b$ = background noise
$_c$ = corrected for background grille
$_g$ = noise
$_o$ = reference
$_s$ = grille noise per sq ft of core area
$_t$ = total noise

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OIL BURNER NOISE

(*Continued from page* 42)

cetane, which is a pure paraffin boiling at 550 F was also similar to No. 2 heating oil.

Of all the tests made, only the lowest boiling paraffins, normal heptane and isooctane, showed noise levels that varied significantly from the other fuels tested. These pure compounds are found in gasolines and are not used as fuels for regulation type oil burners. The low boiling benzene, an aromatic, had the same noise level as No. 2 heating oil. These test results are shown graphically in Fig. 4.

Even here, however, the differences in noise level were not large except towards the higher frequency end of the spectrum.

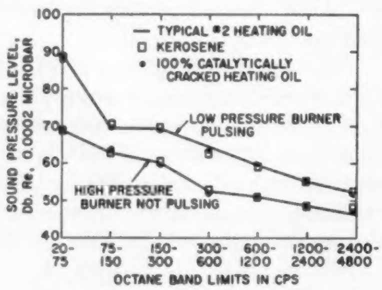
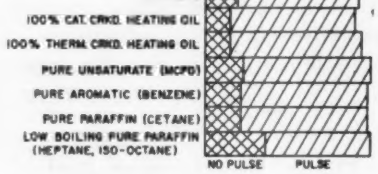


Fig. 3 Sound-pressure-gradients for three fuels

Fig. 4 Test results with various fuels



This can be seen in Table II which contains all the data. While no verified reason is known for this somewhat greater noise level, it may be associated with higher volatility of these compounds which, in turn, means the fuel burns faster in the combustion zone when they are introduced there.

Our conclusion – oil composition does not affect burner noise – We conclude that the oil burners used in home heating equipment are not sensitive to fuel composition from a noise level point of view. The extreme ranges of compositions tested are far beyond the limits which any refinery product could cover. Even the extremes, however, gave essentially the same noise levels. Changing fuel composition, therefore, does not appear to be a method by which oil burner noise can be reduced.

TABLE II

COMBUSTION NOISE FROM VARIOUS FUELS

Sound Pressure Level in Each Octave Band, DB RE 0.0002 Microbars

	Overall (20-20K)	20-75	75-150	150-300	300-600	600-1200	1200-2400	2400-4800	4800-9600	Remarks
No. 2 Heating Oil	89.0	89.0	69.0	69.0	64.5	59.5	55.5	52.0	40.0	Low pressure burner with pulsing flame.
Kerosene	88.5	88.5	70.0	70.0	63.0	59.0	55.0	53.0	41.0	
100% Cat. Cracked Heating Oil	88.0	88.0	69.0	68.0	63.5	59.0	55.0	52.0	40.5	
100% Thermally Cracked Heating Oil	89.0	89.0	69.0	68.5	64.0	59.5	55.0	52.0	40.0	
Methyl Dicyclopentadiene	90.5	90.5	71.0	69.5	66.0	62.0	56.5	50.5	40.0	Low pressure burner with pulsing flame.
Benzene	90.0	90.0	70.5	69.5	65.5	61.5	55.5	50.0	39.0	
n-Cetane	90.0	90.0	70.5	71.5	66.0	62.0	—	—	—	
n-Heptane	90.0	90.0	76.0	73.0	67.0	64.0	60.0	57.5	48.0	
Isooctane	90.5	90.5	72.5	72.5	65.5	62.5	58.5	54.0	44.0	
No. 2 Heating Oil	69.0	67.5	62.0	58.0	52.0	51.0	48.5	47.0	40.5	High pressure burner with nonpulsing flame.
100% Cat. Cracked Heating Oil	69.0	67.5	63.0	57.5	51.5	51.0	48.5	48.0	41.0	
Kerosene	69.0	68.0	62.5	58.0	52.5	51.5	48.5	48.5	40.0	

Know your ASHRAE

ARTHUR J. HESS

As your new President, even though slightly tarnished from use, I wish to express my thanks to you for your confidence in my ability to do the work of this highest office at your command. I also wish to express our thanks to Cecil Boling who preceded me in this office and performed a Herculean task during the very trying period of transition.

During the period of operation of my term of office as President, it is my desire that this President's Page be used to acquaint the membership with the internal workings of the Society. I have been really surprised as I go about the country in the lack of knowledge our membership has as to how the Society functions, and this applies equally to the members from either of the parent groups. I realize that readers of this page are few and far between but perhaps the few who do read it can assist in spreading the story when possible and I hope some Chapter publication can pick it up in some form. Hence, during this time you will hear from each of the Vice Presidents and the Treasurer who will explain to you some phase of Society operation or something about the Society they desire to talk to you about.

Our National organization is made up of the Council, the elected Officers, the Committees and the Staff. These naturally divide into two groups, on the one hand, the Council, Officers and Committees who are elected or appointed by your elected representatives. Generally their job is to set the policies of operation of the Society under the Charter and By-Laws, in other words, to see to it that the Society does what it is supposed to do. As you know, these are dedicated members who serve with no remuneration and in many cases without reimbursement of travel and other expenses. These men will devote all the time they can to do their job but should not be asked to do any more than they can give without too great a sacrifice.

The Staff, on the other hand, is hired by the Society to carry on the day to day business functions of the Society together with implementing and carrying out the policies as set up by the Council, Officers and Committee group. The Staff is also made up of a dedicated group, they must be, to put up with some of our ideas of operation. The Staff is somewhat vulnerable too since they "just work there."

I think it is important that the members know these things so that their approaches to National problems or National contacts can be properly made and so they will have a measuring rod for checking on certain things that are done or how they are done.

I hope each of you has a wonderful vacation.

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W. A. Grant
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Others are saying—

that . . . . . . use of solar energy as a water heating method has proved successful in tropical and sub-tropical areas. An installation cited consists of a number of flat, tank-type absorber panels, made of 20-gauge copper sheet (aluminum may be substituted) enclosed in a wood case with a layer of 1 in. thick glass wool under the panel and a sheet of 1.4-in. glass suspended over it, allowing an air space between the metal and the glass, through which the water is circulated. From these absorbers, flow and return pipes lead to a storage cylinder to which immersion heaters are attached for use when there is insufficient sunlight. *Journal of the Institution of Heating and Ventilating Engineers, May 1959, p 54. (British)*

that . . . . . . experiments on the freezing of meat and fish in liquid nitrogen at a temperature ranging from —195 to —18 C, and subsequent storage in this medium for two weeks showed that the protein solubility remained unchanged, the hydrophilic properties of the meat and fish as evidenced by the drip on thawing and centrifuging and cooking in steam changed only slightly, the sarcolemma was not destroyed but cleavage of the muscle fibers took place, and no changes occurred in the organoleptic characteristics. *Refrigeration Techniques (Xolodil'naya Technika), January-February 1959, p 52. (Russian)*

that . . . . . . considerations the architect and home buyer should specify for a heating and cooling system are: constant and uniform temperatures throughout all rooms, winter air temperatures being maintained within two degrees of thermostat setting and summer air temperatures within three degrees; temperatures above the floor surface in winter not falling below 70 F; variation in temperature between floor and ceiling not exceeding one degree for each 15 degrees difference between indoor and outdoor temperatures in winter, and three degrees for each 10 degrees difference in summer; filtered air should be delivered to all rooms; through the use of a humidifier during the winter humidified air can be added to all rooms; the air conditioning unit is capable of reducing the relative humidity to 50% or less, provided the structure is sufficiently moisture-proof and there are no major sources of moisture within it. *Architectural Record, Mid-May 1959, p 21.*

that . . . . . . thermal environment is becoming an important consideration in schoolhouse planning, particularly with regard to summer cooling, a need created by proposed extension of the school year and increase in summer school enrollments. School administrators are obliged often to approve or overrule their architects and engineers on decisions involving major expenditures on heating, ventilating, fuels, structural materials, and such, all of which have an effect on the school's thermal environment, as well as on its operating cost. A comprehensive five-fold program is manifested in an insert in this publication. *Nation's Schools, May 1959, p 85.*

that . . . . . . polyurethane foam has been introduced as the insulation medium in one of the holds of the Ionic, a refrigerated liner of the Shaw Savill firm. Capacity for refrigerated and chilled cargo is over 400,000 cu. ft. *Modern Refrigeration, May 1959, p 448.*

that . . . . . . at low frequencies the air induction fan is the chief source of cooling tower noise; water splash predominating as a noise-producing factor only at high frequencies. A correlation exists between the overall sound level and the power of the cooling tower fans, and since the sound

(*Continued on page* 103)

RESEARCH PAGE

New solar research data on windows and draperies

BURGESS H. JENNINGS
NECATI OZISIK LESTER F. SCHUTRUM

For more than six months the Research Laboratory has been carrying out a research program dealing with the effectiveness of draperies at windows for preventing or reducing Solar Heat Gain. The work has now been completed and a paper entitled "Solar Heat Gain Factors For Windows With Draperies" has been prepared by Authors Ozisik and Schutrum who shared the research. Intense interest in the results of this investigation has been evidenced by the number of inquiries which have come to the laboratory from consulting engineers and manufacturers, so much so, that it was decided to deviate from previous practice and release a summary of the paper at this time. It happens that this is possible, in the present case, as the essential content can be given by an equation and a series of graphs. This immediate release seems to be desirable also, because in complete form, the paper may not be available until the Transactions appear next year.

In the paper, an equation was prepared by means of which it is possible to determine the total heat being transferred through a glass and drapery combination. A particularly important part of this investigation was the determination of the solar heat-transfer factors K_D and K_d. These factors can be found with the help of Fig. 1 or Fig. 2. When the solar absorptance and transmittance of the material is known, Fig. 1 can be used to find the K_D value for direct solar radiation. When, however, reflectance only is known, Fig. 2 can give the K_D value to within an error of not more than 10 per cent. The K_d value for diffuse solar radiation can be found from the graphs by making use of the corrective note indicated at the bottom of the graphs.

The total heat gain in Btu per hour through a glass and drapery combination can be found by the following equation:

$$Q = K_D I_D A (1-P) + K_d I_d A + U (t_o - t_i) A$$

K_D = Solar heat-transfer factor, dimensionless (From Fig. 1 or 2)

I_D = Intensity of direct solar radiation on a vertical wall, Btu/(hr) (sq ft). (Enter Table 4, Ch. 13, 1959 Guide with the altitude angle from Table 6. Select the direct normal radiation value and multiply it by the cosine of the incident angle from Table 5.)

A = Area of glass surface, sq ft

P = Fraction of glass surface in shade, dimensionless

K_d = Solar heat-transfer factor for diffuse solar radiation, dimensionless. (10% less than the Fig. 1 or 2 values.)

I_d = Intensity of diffuse solar radiation on a vertical wall, Btu/(hr) (sq ft). (Use appropriate value from Table 4, Ch. 13, 1959 Guide.)

U = Over-all coefficient of heat transfer, Btu/(hr) (sq ft) (F). (For a drapery with glass it can be taken as 0.76.)

t_o = Outside temperature, F

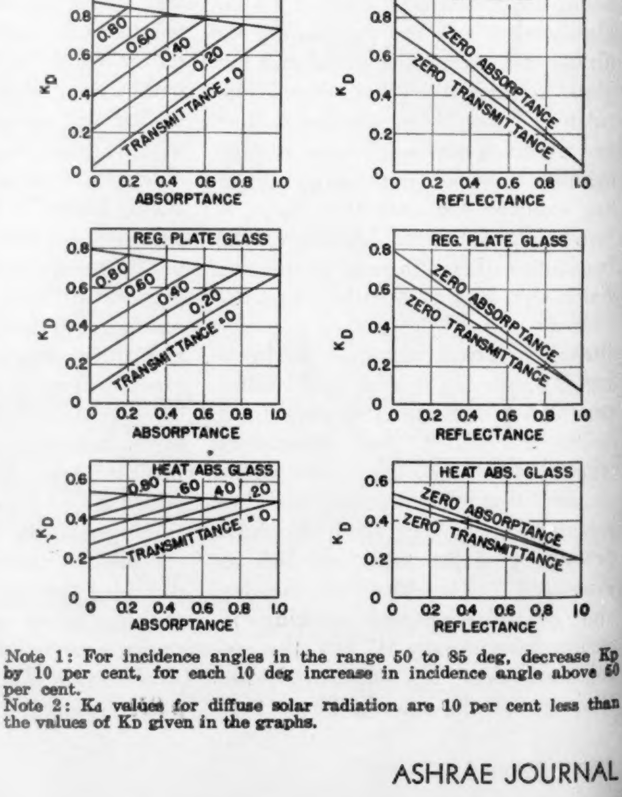
t_i = Inside temperature, F

In order to provide representative information concerning drapery materials and glasses, the authors have included the data of Table I. Values in this table are for normal incidence and include the representative materials that were employed in the test program.

TABLE I SOLAR PROPERTIES OF DRAPERY MATERIALS AND GLASSES
(Determined for Normal Incidence)

Material	Transmittance	Absorptance	Reflectance
Cotton; dark green, 6.06 oz per sq yd	0.05	0.80	0.15
Cotton; dark green, vinyl coated	0	0.85	0.15
Cotton; Beige, 6.18 oz per sq yd	0.23	0.26	0.51
Glass fiber; dark grey	0.11	0.60	0.29
Glass fiber; tan, 7.55 oz per sq yd	0.14	0.44	0.42
Glass fiber; light grey	0.23	0.30	0.47
Glass fiber; off white, 5.72 oz per sq yd	0.35	0.05	0.60
Glass cloth; white, gold striped	0.54	0.05	0.41
Dacron; white, 1.8 oz per sq yd	0.70	0.02	0.28
Glasses;			
Ordinary Window	0.87	0.05	0.08
Regular plate	0.77	0.16	0.07
Heat absorbing	0.41	0.53	0.06

Figs. 1 and 2 Values of the solar heat-transfer factor K_D for direct solar radiation and incidence angles ranging from 0 to 50 deg



Note 1: For incidence angles in the range 50 to 85 deg, decrease K_D by 10 per cent, for each 10 deg increase in incidence angle above 50 per cent.
Note 2: K_d values for diffuse solar radiation are 10 per cent less than the values of K_D given in the graphs.

Burgess H. Jennings is Research Director, Necati Ozisik is Research Engineer and Lester F. Schutrum is Research Supervisor of the ASHRAE Research Laboratory.

Education—

our inexhaustible resource

EARL L. BUTZ

Living standards unparalleled in any other country on the face of the earth have been brought for all our people by the American economic and political system. One need only go abroad for a short time to any part of the globe to see convincing proof that this is true.

The economic growth of our nation over the past couple of decades has been truly phenomenal. Our economy has doubled in size in less than 20 years. Reliable predictions are that it will double again the next 20 years. Real income per person in America has increased by about 50 per cent in the last 20 years. Indications are that this likewise will increase about 50 per cent in the next 20 years.

America is on the threshold of its most challenging and most prosperous decade. This is the age of science and technology, based on research. The frontiers of the mind have replaced the frontiers of geography. A thrilling experience awaits every citizen in this great land who has the capacity and the imagination to dream constructively.

Since the geographic frontier in America is gone, no longer can a young man "Go West" and stake out his claim. But the scientific frontier in America is barely scratched. And the scientific frontier has no effective limit. It is limited only by the mind and the imagination of man. Its horizons are vertical, not horizontal. Organized and imaginative research is the vehicle which will push the scientific frontier beyond limits we scarcely dare dream of today.

The most constant thing about our society is change. These are days when research and industry leaders need greater vision than ever before. They need to spend more time in constructive dreaming than many of us have the inclination or time to do. Never before in history has the future been so near to us as now. Research and education are shortening the time span of progress. We shall occupy ringside seats in scientific and technological developments during the next ten years equivalent to changes which our fathers took a generation to accomplish.

We must remind ourselves constantly that the advances of science can be applied most effectively by individuals in a free industry, unhampered by excessive governmental regulation and restriction. Freedom of choice is a basic pillar of our economic system.

In our free society, the right to succeed carries with it the right to fail. If we remove the right to fail, we ultimately will also remove the right to succeed beyond mediocrity. Men of vision and ambition want an environment which guarantees freedom to choose, freedom to experiment, freedom to become more efficient, freedom to seek and develop new markets, freedom to dream, and freedom to enjoy economic rewards if their dream is successful.

Keep government the servant of all of us—not our master.

The Great American Dream, emerging with breathtaking rapidity, can be reduced to a five term formula of: (1) Abundant and varied natural resources; (2) a population of 175 million persons with a great deal of hybrid vigor in it; (3) a form of government in which every individual has an opportunity to go just as far as he wants; (4) a code of ethics based on the Judaeo-Christian tradition; and (5) a program of universal education and personal development.

The first four of these items are relatively fixed. They cannot be changed much or even fast, but the fifth—the development of education and leadership—is really expandable. I would emphasize that area.

We talk a lot in America about horsepower. Our greatest national resource is brainpower. The complex scientific and social environment in which we live demands increasingly competent men and women. Let us develop the brainpower of young America in such a way that generations to come can enjoy fully the technological and social developments which await us.

Change is the law of progress. The challenge which faces us is to direct the change along constructive and beneficial channels. Our problem becomes one of intelligent analysis and direction of the future.

None of us would want to live under the economic institutions which prevailed just a short 20 years ago. Yet I am confident that some of you, 20 years ago, vigorously resisted changes which were then occurring. As we look back now, we wonder why.

Likewise, 20 years from now we shall look back on 1958 to be a little amused that we were so fearful of change, rather than directing our energies toward channeling change down beneficial pathways. The future belongs to those who prepare for it.

Dean of Agriculture Earl L. Butz of Purdue University in part thus addressed the 45th Semiannual Meeting of ASRE at the Welcome Luncheon, December 1, 1958, under the designation of "The Great American Dream".

Meetings ahead

July 13-16—United States Air Force, Air Conditioning and Refrigeration Conference, Dayton, Ohio.

August 19-26—10th International Congress of Refrigeration, Copenhagen, Denmark.

September 2-4—Cryogenic Engineering Conference, University of California, Berkeley, Calif.

September 25-29—American Meat Institute, Palmer House, Chicago, Ill.

October 5-7—American Gas Association, Annual Convention, Chicago, Ill.

October 30-November 2 – Refrigeration Service Engineers Society, Annual Convention, Atlantic City, N. J.

November 2-5—11th Exposition of the Air-Conditioning and Refrigeration Industry, Atlantic City, N. J.

November 9-13 – National Electrical Manufacturers Association, Annual Meeting, Atlantic City, N. J.

November 17-19 – Building Research Institute Conference, Washington, D. C.

December 3-5 – National Warm Air Heating and Air Conditioning Association, Annual Convention, St. Louis, Mo.

December 26-31 – American Association for the Advancement of Science, Annual Meeting, Chicago, Ill.

February 1-4 – American Society of Heating, Refrigerating and Air-Conditioning Engineers, Semi Annual Meeting, Dallas, Texas.

February 1-4—2nd Southwest Heating and Air-Conditioning Exposition, Dallas, Texas.

March 6-10 – National Association of Frozen Food Packers, 19th Annual Convention-Exposition, Chicago, Ill.

May 1-4—Air-Conditioning and Refrigeration Institute, Annual Meeting, Hot Springs, Va.

June 13-15—American Society of Heating, Refrigerating and Air-Conditioning Engineers, 67th Annual Meeting, Vancouver, B. C.

People

Albert F. Metzger, Allegheny County Steam Heating Company, was recently elected 1st Vice President of the National District Heating Association. Third Vice President chosen at the annual meeting was **James C. Thompson,** Georgia Power Company. **Ernest T. Smith,** American Gilsonite Company, was named to the Board of Directors.

Robert W. Ayling has been named Engineering Section Manager, large compressors, at the Westinghouse Electric Corporation's air conditioning div. An MIT graduate, he comes to his present position from Carrier Corporation where he was manager of compressor development for the Unitary Equipment Div. Most of his career has been in the development of refrigerant compressors; he holds seven U. S. patents.

Howard Crusey returns to Drayer-Hanson, Div of National-U. S. Radiator Corporation as Sales Engineer in charge of Sales Agent groups in several territories. For the past year he has been on special missile facility design projects.

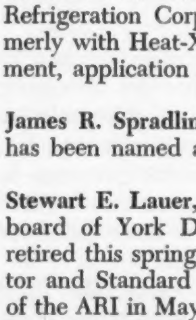
Robert L. Davidson, Vice President of Kurz-Kasch, Inc., is the newly-elected Vice President and Director of the Society of the Plastics Industry, Inc.

Louis L. Narowetz fell to his death from the 22nd floor of a Chicago building in May. A mechanical engineer, he was owner of Narowetz Heating and Ventilating Company, although since 1951 had been in the process of turning over the stock to employees. Early in 1959, he was elected a Life Member of ASHRAE.

Russell E. Comstock has been appointed District Representative for Copeland Refrigeration Corporation, with headquarters in the southeast. He was formerly with Heat-X and Brunner Manufacturing Company in product development, application and sales activities.

James R. Spradling, a student at the University of Missouri School of Law, has been named a director of the Carthage Ice and Cold Storage Company.

Stewart E. Lauer, former President of York Corporation and chairman of the board of York Div, Borg-Warner Corporation, and **K. B. Thorndike,** who retired this spring as Vice President of Detroit Controls Div, American Radiator and Standard Sanitary Corporation, were honored at the annual meeting of the ARI in May. Both were given engraved silver bowls reading, "Presented by the Air-Conditioning and Refrigeration Institute, in Grateful Appreciation of Long Years of Service to the Industry."



Paul Komroff is now Vice President in Charge of Air Conditioning Engineering for the Consumer Products Div of Emerson Radio and Phonograph Corporation. An alumnus of Yale University and Brooklyn Polytechnic Institute, he was in charge of Engineering for the Quiet-Heet Manufacturing Company at the time it became a subsidiary of Emerson. A co-author of several patents, he has been active in the design and improvement of room air conditioning models.

Harold A. Lockhart, Vice President in charge of engineering, Bell & Gossett Company, was honored in June for his 25 years of service with the company. He was presented with a gold watch and a 25-year service pin. He joined the company in 1934 as a member of the engineering staff, and was elected a Vice President and member of the board of directors in 1953.

Ralph G. Owens, Dean of Engineering at the Illinois Institute of Technology, has been elected Vice President for general and regional activities east of the Mississippi by the American Society for Engineering Education. His term will run from 1959 to 1961.

Warren H. Scott is now Product Manager for residential air conditioning and low-voltage thermostats in his new post with the Temperature Controls Group of Minneapolis-Honeywell Regulator Company. At one time in charge of technical investigation at the Austin (Texas) Air Conditioned Village, he has held a variety of engineering positions, including Test Engineer, Assistant to the Service Manager, and residential air conditioning Sales Engineer.

Bernard E. Proctor, Head of the Food Technology Department, Massachusetts Institute of Technology, will serve on the newly formed Advisory Editorial Board of the AVI Publishing Company. His duties will include advising the company concerning the scientific and technological fields in which books are needed, as well as suggesting authors best suited to write manuscripts.

Herbert P. Tinning, former ASRE Assistant Secretary in charge of Membership and Sections, transfers from the New York office of Dunham-Bush Inc. to the New Jersey area, where he will cover sales.

Harold A. Mosher, Assistant Director of Engineering, Eastman Kodak Company, has been elected president of the National Society of Professional Engineers. He is currently serving a term as the vice president for the northeastern region. A graduate of Occidental College at Los Angeles, he has been associated with Bell Telephone Laboratories and Consolidated Film Industries.

Charles W. Hanson and **Ralph H. Stebbings** join to form Hanson & Stebbings Company, representing Recold Corporation refrigeration and air conditioning products in Florida.

Chester B. Morrison, deceased, had only recently retired as Vice President of the York Div of Borg-Warner International Corporation.

J. C. Stangl has been appointed General Sales Manager, Western Div of Anthes-Imperial Company Ltd. He has been Manager, plumbing and heating sales, with the same division.

Donald F. Williams, with the title of Chief Cabinet Engineer, will supervise all cabinet design for Amana Refrigeration, Inc. A graduate of Iowa State College with a Bachelor of Science degree, he was recently associated with Westinghouse Electric Corporation as project engineer in product design. Mr. Williams was also with Amana from 1952 to 1955. He is a registered professional engineer in the state of Iowa.

A. K. Johnson has been appointed Cleveland manager of Ilg Electric Ventilating Company. He has been with the organization since 1947, serving two years in the Chicago sales office and ten years in the Milwaukee office.

Olan J. Hill will cover the state of Kentucky as Sales Engineer for Acme Industries, Inc. He is a past president of the Bluegrass Chapter of the former ASHAE.

D. T. Stikeleather of Cincinnati and **J. P. Ashcraft** of Dallas have been named sales representatives for McQuay refrigeration products.

A. J. Lane, District Manager (Atlantic Central area) for Dole Refrigerating Company, was elected International Director of the Refrigeration Service Engineers Society. In his position with the society he will represent the Carolinas, Georgia and Florida.

Robert W. Carvell advances to the position of Sales Promotion and Advertising Manager, Henry Valve Company. For thirteen years he has been closely associated with sales in the refrigeration and air conditioning industry, as Sales Manager of Bohn Aluminum Company's Betz Div, on wholesaler activities, with the Kennard Corporation as Assistant Sales Manager, and as Field Engineer for Alco Valve Company.

BULLETINS

Water in Air Conditioning Systems. In question and answer form, this flyer includes discussions of corrosion and scale prevention, problems in closed circulating systems, and control of slime and algae. Bulletin AQ-59.
Water Service Laboratories, Inc., 615 West 131st St., New York 27, N. Y.

Voltage Regulators. Application, description, selection, weights, dimensions and connection diagrams are given in this 60-page illustrated booklet, GEC 1450A, describing this company's redesign of dry type and liquid filled induction voltage regulators of 60 and 400 cycles. Dry type single-phase units are offered in ratings up to 180 kva, 600 volt and below, three-phase up to 720 kva for industrial and electronic applications, with manual, motor-driven, or automatic control. Liquid filled units are rated up to 2,200 kva, 13,800 volt and below.
General Electric Company, Pittsfield, Mass.

Weatherstripped Door Stop. Form SDS-1 is a flyer describing the Seal-Draft Weatherstripped Door Stop, a vinyl sealing gasket imbedded into a door-stop-shaped extrusion which attaches to the door jamb.
Seal-Draft Div, Sun Screen Products, Inc., 2220 North Division St., Spokane 21, Wash.

Sludge Solvent. Specially formulated liquid that eliminates, by chemical means, many of the difficulties encountered in the use and handling of residual fuel oil, is covered in 4-page Bulletin 594. The fuel oil sludge solvent is made to disperse and remove sludge present in tanks, suction lines, strainers and other parts of the fuel oil systems. It is also useful in preventing new accumulations of sludge.
Betz Laboratories, Inc., Gillingham & Worth Sts., Philadelphia 24, Pa.

Calcium Chloride Brine. Suggestions for accurately testing brine concentration and advice on strengthening brine are given in this 4-page bulletin. Brief RB-1, prepared especially for refrigerating engineers. Included are data on tests for ammonia leakage, alkalinity and acidity of brine, corrosion inhibitors, methods for cor-

(Continued on page 99)

Life Members

In recognition of distinguished service to the Society and to the industry, a qualified member in good standing for thirty years and over the age of 65 is honored by the grade of Life Member, carrying all the rights and privileges of his former membership grade as well as exemption from further dues.

Receiving their Life Member Certificates recently were:

Otto W. Armspach, who recently retired from Carrier Corporation, where he was in the Application Engineering Dept. Educated at the Armour Institute of Technology, he has held positions with the Chicago Department of Health, the U. S. Bureau of Mines, Kroeschell Engineering Company (Vice President and Chief Engineer), and the ASHAE Research Laboratory, among others. He is the author of a series of articles on theatre comfort.

A. B. Algren, Professor and Head, Div of Heating, Ventilating, Air Conditioning and Refrigeration, University of Minnesota, who except for World War II has been with the university since 1927. During the war he was Regional Chief of Training on the War Manpower Commission. He has served the Society most prominently in the research and publications areas, and in January of this year, completed the three year term as member of Council. For the past four years he has conducted a cooperative research project with the Society on Air Filtration.

Harold E. Adams, Vice President of Engineering, Nash Engineering Company, active in the formation of the Connecticut Chapter of ASHRAE, and author of numerous technical papers. Mr. Adams began his career with the Lake Torpedo Boat Company, after graduating from Middlebury College with a BSME degree. He then moved to the position of Mechanical Engineer with the Bridgeport Engineering Company, and in 1924 joined his present firm. He became a Vice President in 1950. He holds over 100 U. S. and foreign patents on original pump and compressor inventions.

J. Earl Seiter, Manager of the District Steam Department, Baltimore Gas and Electric Company. A charter member of the Baltimore Chapter, he has been active on the ASHAE Guide. Mr. Seiter is also a past president and honorary member of the National District Heating Association. A co-author of **Principles of Economical Heating,** he was instrumental in the publication of **District Heating Handbook, Third Edition.**

Lester S. O'Bannon, Professor in the Mechanical Engineering Department, A and M College of Texas. Active in both industry and research, this new Life Member has spent most of his career teaching: at his alma mater, the University of Kentucky, where he was head of the Mechanical Engineering Department; as Visiting Professor at the University of Michigan; and now at A and M. While in Kentucky, he served as Consulting Engineer. He has written several research papers and other articles.

Ralph P. Cook, who retired in January after many years with Eastman Kodak Company. He began there as a mechanical engineer with a master's degree from the University of Michigan, and for a number of years specialized in air conditioning, drying and solvent recovery problems. In 1936 he became Superintendent of the Engineering Div, and in 1953, Director of Engineering. He has served on the Guide and publication committees of the Society.

Herman Worsham, due to retire soon from the Frigidaire Div, General Motors Corporation, after 25 years there. Upon receiving his Master's Degree in Mechanical Engineering from the University of Kentucky, he joined Carrier Corporation where he remained for 16 years in engineering and sales. He has spent most of his time with General Motors in Dayton headquarters where he is now Supervisor of Government Sales. During World War II, Mr. Worsham was a member of their War Products Training Section.

George A. Teeling, President of Utilities Engineering Company, Inc., and partner of Teeling & Spindler, Consulting Engineers. At one time a member of the Isthmus Canal commission as a designing engineer, he worked with B. F. Sturtevant Company in heating and ventilation work until 1931 when he began consulting work in heating, air conditioning, steam, power and industrial work.

Lester R. Ries, retired from active work in 1955, and currently doing consulting and technical writing. At different times in his career, he served the University of Chicago and Oberlin College on their physical plant staffs. A past president and member of the board of governors of ASHRAE's Illinois chapter, he has also served the Northern Ohio group. In 1950 he completed a two year term on the Admissions and Advancement Committee.

(Continued on next page)

Jean Paul Leinroth, Manager, Industrial and Commercial Sales, Public Service Electric and Gas Company, Newark, N. J. A graduate of Cornell University in 1912 with an ME degree, he entered the gas field as a cadet engineer with the United Gas Improvement Company. Since then he has been active in this line of work and includes in his activities a year spent at the University of Illinois, College of Engineering, as an instructor.

Harold Clark, manufacturers representative specializing in heating, ventilating and air conditioning equipment.

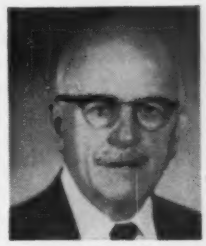
A 1916 graduate of the College of Mechanical and Electrical Engineering, University of Kentucky, he first worked with the Buffalo Forge Company. From 1920 to 1925, he was with the American Blower Company, leaving this to begin his own business. From 1928 to 1932 he served as secretary to the Michigan chapter, ASHRAE.

Joseph C. Fitts, former secretary of the Mechanical Contractors Association, received their Distinguished Service Award in 1957. He was cited for his 38 years of service, during which successful industry-wide standards were set. Since his retirement, he has continued as consultant to the national association. A past chairman and currently member of the ASHRAE Charter and By-laws Committee, he is also on the Guide Advisory Committee.

L. L. McConachie, native Canadian, who entered the heating and plumbing business in Detroit in 1920 as a partner in the McConachie and Reid Company. He is presently chairman of the board of what is now the L. L. McConachie Company. President of the local Heating, Piping and Air-conditioning Contractors Association for twelve years, and member of the national board for thirteen, he was elected national president of that association in 1949, for a one-year term. He holds a diploma from the William Hood Dunwoody Institute in Minneapolis.

Clarence N. Warren, who has served over fifty years with Hayes Brothers, Inc. While working as tracer and draftsman in an architectural office, in 1906, he met Joseph G. Hayes (late Life Member) who asked him to join his firm and begin an engineering department. From 1919 until his retirement in October 1956, Mr. Warren served as Vice President and Director of that firm. One of the earliest Professional Engineer's certificates, No. 159, was issued to him in 1921.

Leo A. Brissette, who joined the Trask Heating Company in Boston in 1899 and moved from positions as stenographer, assistant bookkeeper and apprentice steamfitter to Treasurer and General Manager. In 1927, he became owner of the business, and since then has been instrumental in the design of many New England and New York private residence heating systems.



GEORGE A. HORNE

1877-1959

George A. Horne, a Past-President of the former American Society of Refrigerating Engineers, and an honorary member of that Society since 1944, died in May, at the age of 82.

As Vice President and Chief Engineer of the Merchants Refrigerating Company, he waged a strenuous campaign throughout his career to improve cold storage practices and develop the field of refrigerated warehousing.

Research work on egg storage (with the late Mary E. Pennington) and pioneer research on the application of ozone to egg storage rooms and humidity control, were two of the many projects in which he engaged.

Authoring a series of technical papers which he contributed to ASRE, he was most noted for articles covering Compound Ammonia Compression, although his range of subject matter included other aspects of refrigeration design as well as the engineer's place in society.

Among the papers appearing in Refrigerating Engineering were a "Theory of Cooling Towers Compared with Results in Practice" (with the late B. H. Coffey); and "Performance of Single-Acting Simple Ammonia Compressor and Tubular Condensers."

"Recent Improvements in Refrigerating Apparatus" (with Fred Ophuls) was presented at the Fourth International Congress of Refrigeration in London, in 1924.

In 1936, Mr. Horne presented his views on Refrigerated Warehouse Operating Costs, at the Seventh International Congress of Refrigeration in the Hague. At this Congress, he was the official United States delegate.

An engineering graduate of Lehigh University, his early work in the field of chemical engineering was in the laboratories of Thomas A. Edison, and as chemist and chief chemist for B. T. Babbitt, soap manufacturers.

In the latter, he had charge of all research and experimental work, developing new formulas and designing a small laboratory vacuum apparatus for the distillation of fatty acids, also a vacuum still for refining glycerin.

In the 1920's, prior to and following his presidency of ASRE, Mr. Horne was active broadly in the Society's interest. He also served on the ASME main Committee for Power Test Codes, and as chairman of the Joint ASME-ASRE subcommittee on Refrigerating Systems.

What ASHRAE Chapters are doing

Biological, physiological and economic aspects as well as the various types of air conditioning gained attention at the late spring meetings. Many combined chapters elected officers and some chose new names for the coming year.

CENTRAL OKLAHOMA . . . Answering questions previously submitted to him by this chapter, Roy J. Thompson discussed specific electrical problems encountered in the heating, refrigeration and air conditioning industries. A partner of Carnahan and Thompson Engineers, the speaker covered such subjects as reduced-voltage starting of motors, unbalanced three-phase primary voltages in residential areas, overload protection of motors, and other related matters. In addition, he received and answered many more questions from the floor.

UTAH (H) . . . There are five school districts receiving school aid from the state of Utah, the Director of the State Building Board told these members in May. Glen Swenson, in reviewing the State Building Board Ten Year Building Plan, commented upon the state's new attitude on "agent-designers" and advised that these "agent-designers" can not now be used on any State job.

MEMPHIS . . . Aims and objectives of the Construction Specifications Institute were up for discussion at the May meeting of this group when Thomsen Guth, an associate of the architectural firm of Walk C. Jones, Jr., planned to address members.

FORT WORTH and DALLAS . . . Illustrations, demonstrations and a miniaturized working model of the dehydration units used in his office augmented the talk on the History and Nomenclature of Modern Refrigerants, presented by Dan Anderson, General Chemical Div, Allied Chemical Company, at the May meeting.

TOLEDO (H) . . . Economic aspects of air conditioning include human efficiency, personnel turnover, competition, customer comfort and interior preservation, said William Ortman at the early May meeting. His talk, Economics of Complete Year Round Air Conditioning, dealt with the growth of air conditioning and the questions asked by owners concerning full and incomplete design of systems from the standpoint of temperature, humidity, cleanliness and distribution.

Pending establishment of unified ASHRAE chapters, meetings of former ASHAE Chapters (H) and former ASRE Sections (R) are indicated above.

SOUTHERN CALIFORNA (H) . . . Robert Barnacutt, engineering consultant for the Bureau of Hospitals, State of California, presented an informative talk at the early May meeting of this group.

REGION X . . . Hosted by the San Francisco chapter, this regional meeting held in mid-May covered three days of activity. On the first day a work shop on Cooling Coil Selection was conducted by T. R. Simonson, Consulting Engineer, D. E. McLeod, Consulting Engineer, and D. D. Wile, Chief Engineer, Recold Corporation and 1st Vice President of ASHRAE.

The second day found Lincoln Bullion, Consulting Engineer, moderating a panel on High Temperature Hot Water and its uses in design. Members in the discussion were: Boilers, E. R. Skequine; Pumps, Harold Lockhart, Bell & Gossett Company; and Controls, E. C. Hallenger, Minneapolis-Honeywell Regulator Company.

Reciprocating Compressors vs. Centrifugal were debated by W. B. Ludwig, York Corporation, who spoke on those below 10 deg; Robert McKee, Pacific Fruit Express, who covered those above 10 deg; and Gerald Yaffe, Worthington Corporation, who talked on Chillers.

That afternoon Joseph Clancy, California State Div of Architecture, headed a panel on High Velocity Systems of Air Distribution. Richard Trit, Tuttle and Bailey Corporation; Norman Janisse, Johnson Control Corporation; and Erwin Gebhart, Western Air and Refrigeration Company, comprised the panel members.

On Saturday, regional chapter meetings were held.

FLORIDA WEST COAST . . . Gas air conditioning equipment was discussed at the June meeting of this group when Harry Phipps, Houston Corporation, addressed members on Natural Gas.

ONTARIO (H) . . . Company publicity policies and ways of dealing with public relations were the subject of Ken Gould's May discourse on the Face on the Corporation Floor.

JOHNSTON . . . Meeting jointly with the local chapter of AIEE in May, these members heard Ray Bell, Business Administrator, C-Stellarator Associates, speak on a joint project of Allis-Chalmers and RCA to build a $35 million device to produce

controlled thermonuclear fusion. The basic theory behind the development was reviewed and the problems to be encountered in building it discussed.

NORTH JERSEY . . . Urging that we consider abandoning conventional comfort standards and design air conditioning on the basis of health requirements, C. P. Yaglou, Harvard University, spoke to these members in May on Physiological and Biophysical Aspects of Air Conditioning.

SOUTHWEST TEXAS (H) . . . Elliot Whitman and Osmund A. Brynie of Fort Sam Houston addressed this chapter in May, giving explanations and descriptive talks on government specifications pertaining to their specific areas of control, especially cooling towers.

COLUMBUS . . . Feature of the May meeting was a Heating and Air Conditioning Fair at which chapter members exhibited equipment, controls and other material of interest to the group.

CENTRAL NEW YORK . . . Charles N. Rink of the Industrial Acoustics Company, Inc., spoke to this group in mid-May. His subject was Noise Control in Air Conditioning Systems.

WISCONSIN (R) . . . On the agenda for the June meeting was a lecture on Product Noise—Its Measurement, Rating and Control. Mr. William M. Ihde, Midwest District Office Manager, General Radio Company, was the invited speaker.

BATON ROUGE (H) . . . Preparing assembled members for the fast approaching cooling season, Walter P. Glancy, Southwest District Manager, Aerofin Corporation, planned to discuss Application Problems with Cooling Coils, at the May meeting.

SOUTHERN ALBERTA . . . Various types of fans and their use in the industry were covered by Robert Junker at the May Meeting. Fan curves were touched upon, showing their selection with system characteristics and method of usage.

FLORIDA WEST COAST . . . Types of plastics pipe, their various properties and typical applications were covered at the May Meeting of this chapter when B. K. Batzer, of Plastiline, Inc., met with members.

PUGET SOUND (H) . . . Speaker at the May past presidents night, Harold Harty, Supervisor of the design development operation, Hanford Laboratories, General Electric Company, planned to talk on the Plutonium Recycle Program and the Test Reactor.

NEW YORK . . . Design Considerations for Large Lithium Bromide Systems was the topic of the mid-May evening when John G. Reid, Jr. met with these men. Among the problems the York Corporation engineer covered were cooling water treatment, noise reduction, lower power requirements per ton of refrigeration, crystalization, control and purge systems.

Organizing under post-merger set-ups for the coming 1959-1960 season, some ASHRAE chapters already have selected their officers. The accompanying record reflects such information as has reached Headquarters. Other changes will be listed in later issues.

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(*Continued on page* 106)

ASHRAE INTERSOCIETY STANDARDS REPRESENTATIVES

A-114 Application Standards for Thermal Insulating Materials
Representative:
M. W. Keyes, Chairman

B-9 Safety Code for Mechanical Refrigeration
Representatives:
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AMERICAN STANDARDS ASSOCIATION PROJECTS (Sponsored by ASHRAE)

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A. J. Hess
W. W. Higham

B-38 Household Refrigerators and Home and Farm Freezers
Representatives:
E. C. McCracken, Chairman
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B-53 Refrigeration Terms and Definitions
Representatives:
G. B. Priester, Chairman
Harold J. Ryan

B-59 Mechanical Refrigeration Installations on Shipboard
Representative:
W. L. Keller

B-60 Methods of Testing for Rating Thermostatic & Constant Pressure Expansion Valves
Representative:
D. C. Albright

Z-74 Fundamentals of Performance of Effluent and Gas Cleaning Equipment
Sponsors:
ASHRAE, ASME (Administrative)
Representative:
K. E. Robinson

A-13 Scheme for Identification of Piping Systems
Representatives:
Harry H. Bond
Crosby Field

A-53 Building Code Requirements for Light and Ventilation
Representative:
John G. Eadie

A-62 Coordination of Dimensions of Building Materials and Equipment
Representative:
A. W. Knecht

B-2 Pipe Threads
Representatives:
S. W. Brown
L. L. Munier

B-16 Standardization of Pipe Flanges and Fittings
Representatives:
C. W. Hudzietz

B-31 Code for Pressure Piping
Representatives:
J. L. Wolf

AMERICAN STANDARDS ASSOCIATION PROJECTS (Not sponsored by ASHRAE)

S. E. Rottmayer (only on Subcommittee for revision of Sec. 5 Refrigerant Piping)

B-40 Indicating Pressure and Vacuum Gages
Representative:
Bernhard Willach

B-72 Dimensional Standards for Plastics Pipe
Representative:
W. J. Olvany

B-76 Industrial Cooling Towers
Representatives:
P. A. Bourquin
John Engalitcheff, Jr.

B-78 Heat Exchangers for Chemical Industry Use
Representative:
C. E. Drake

C-85 Terminology for Automatic Controls
Representatives:
C. H. Burkhardt
K. B. Thorndike

K-61 Storage and Handling of Anhydrous Ammonia and Ammonia Solution
Representative:
C. F. Holske

S-1 Physical Acoustics
Representative:
J. B. Chaddock

Y-1 Abbreviations
Representative:
N. N. Wolpert
Alternates:
R. W. Roose
C. H. Flink

Y-10 Letter Symbols
Representative:
B. E. Short
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C. H. Flink

Y-14 Drawings and Drafting Room Practice
Representatives:
F. Honerkamp
E. R. Wolfert
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Y-32 Graphical Symbols and Designations
Representatives:
E. H. Munier
E. R. Wolfert
Alternate:
C. H. Flink

Z-9 Safety Code for Exhaust Systems
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W. M. Wallace, II
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W. S. Bondy

Z-11 Petroleum Products and Lubrication
Representatives:
B. L. Evans
W. J. Simpson

Z-17 Preferred Numbers
Representative:
D. J. Renwick

Z-48 Method for Marking Portable Compressed Gas Containers to Identify the Material Contained
Representative:
Herbert Wolf

Z-62 Uniform Industrial Hygiene Standards
Representative:
A. D. Brandt

NATIONAL FIRE PROTECTION ASSOCIATION

56 Hospital Operating Rooms
Representative:
R. P. Gaulin
Alternate:
N. Glickman

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

PTC-23 Atmospheric Water Cooling
Representative:
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PTC-25 Safety and Relief Valves
Representative:
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AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Subcommittee on the Evaluation of Hermetic Motor Insulation
Representatives:
R. T. Divers
R. C. Burns

NATIONAL ASSOCIATION OF CORROSION ENGINEERS

Intersociety Corrosion Committee
Representatives:
F. N. Speller
B. A. Phillips

AMERICAN SOCIETY FOR TESTING MATERIALS

A-5 Corrosion of Iron and Steel
Representative:
F. E. Medon

B-3 Corrosion of Non-Ferrous Metals and Alloys
Representative:
R. C. McHarness

C-16 Thermal Insulating Materials
Representatives:
C. F. Kayan
F. B. Rowley
Alternate: **E. R. Queer**
This committee has a subcommittee known as Planning Committee on Thermal Insulating Materials
Representative:
E. R. Queer

D-2 Petroleum Products and Lubricants
Representative:
E. S. Ross

D-3 Gaseous Fuels
Representative:
E. A. Norman, Jr.

D-19 Industrial Water
Representative:
D. R. Walser

D-22 Methods of Atmospheric Sampling and Analysis
Representative:
M. G. Kershaw

AIR-CONDITIONING AND REFRIGERATION INSTITUTE

ASHRAE - ARI Standards Liaison Committee
Representatives:
P. W. Wyckoff
A. T. Boggs, III

AMERICAN MEAT INSTITUTE

ASHRAE-AMI Meat Packing
Representatives:
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B. C. McKenna
C. D. Macy
F. P. Neff
K. E. Wolcott

CANADIAN STANDARDS ASSOCIATION

Safety Code for Hospital Hazards
Representative:
J. Klassen

Intersociety standards

A. T. BOGGS III

ASHRAE Technical Secretary

Intersociety Standards: The principal responsibility of the Standards Committee is the development of standards as required by the industries represented in the Society. There are several organizations allied with ASHRAE that initiate or promulgate standards of interest to ASHRAE members. These intersociety standards receive consideration by ASHRAE through appointed representatives.

At the present time ASHRAE is represented by 84 representatives on 45 intersociety standards committees. These committees and the standards they are developing are published in this issue on page 80. It will be noted that the majority of these intersociety standards are within the framework of ASA. Of the 45 committees, 29 are ASA Sectional Committees.

It is the responsibility of these representatives to see that standards or reports published by the respective intersociety committees do not conflict with established ASHRAE standards. They also present the Society viewpoint in the development of standards related to the fields of interest in the Society. Before an intersociety standard is published in its final form, the Standards Committee must consider it and indicate to our representative the manner in which to vote.

New York City Rulings: Revised rules concerning the installation of ventilating and air conditioning systems in the City of New York became effective June 1, 1959. These rules apply to any ventilation or air conditioning system installed or altered after that date.

The rules are principally concerned with the requirements of building construction and installation of this type of equipment. Included also are requirements relative to the necessary applications and permits which must be filed prior to the installation or alteration of such equipment. A major part of these rules is concerned with construction details with particular emphasis on construction, support, lining, and termination of ducts. For those who may be required to comply with such rules and regulations, copies are available from the New York City Clerk.

U L Proposed Standard: A proposed standard for heat pumps has been distributed by U L for review and comment. With the exception of certain heat pumps which may be affected by the new requirements included in the proposed standard, it is anticipated that no changes in listed heat pumps will be required by the adoption of this standard.

One new requirement includes the Input Test. Input requirements for the cooling cycle are identical to those for air conditioning, central cooling. The input for the heating cycle is based on air and water temperature tests and static pressure tests. This part of the requirement states that "Electrical components and combustible material within a heat pump or combustible structure should not be overheated. Electric and thermal insulation shall not be impaired. Refrigerant-containing parts shall not be subject to excessive pressure and the refrigerant shall not be released under the stated test conditions."

A series of eight tests are established as follows: 1. Input; 2. Continuity of Operation; 3. Limit Control Cut Out, and the following temperature and pressure tests; 4. Heating Operation; 5. Restricted Inlet, Fan Failure, Blocked Outlet; 6. Heating Operation; 7. Restricted Inlet, Fan Failure, Blocked Outlet; 8. Cooling Operation. A defrost test is also included and requires that a defrost system shall not cause excessive temperatures on wiring or electrical components or excessive pressures in the heat pump during a normal defrost cycle.

It also requires that a defrost system shall not ignite combustible material or cause the emission of flame or molten metal or release refrigerant from the system if the shut-off control fails. Other parts of the requirement concern strength tests of refrigerant-containing parts, rupture member tests, and a fusible plug test.

MHMA: Standards for heating, electrical, and plumbing systems of trailers have been published by the Mobile Homes Manufacturers Association. These standards which include procedures for construction of mobile homes, including the design of distribution systems for heating, ventilation, and air conditioning, have been submitted to ASA for consideration as American Standards.

A conference on this subject has been called by ASA for July 7 at which the subject will be discussed. ASHRAE will be represented at this conference and a more complete report will be made at a later date.

Candidates for ASHRAE Membership

Following is a list of 105 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before July 31, 1959 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted upon by the Board of Directors.

Note: * Advancement † Reinstatement

REGION I

Massachusetts

BURKE, B. S., Chief Engr., Second Bank-State Street Trust Co., Boston.

MACWALTER, ANDREW, Asst. Engr., Bethlehem Steel Co., Quincy.

PISTORINO, A. L., Sr. Mech. Engr., Anderson-Nichols & Co., Boston.

USSIA, SAMUEL, Gen. Engr., Consulting Engineers Inc., Boston.

New Jersey

DEFABRITES, GUY, Maint. Foreman, Radio Corporation of America, Camden.

FRANCIS, W. G., Engr., Hudik-Ross, Inc., Hackensack.

LESCHINSKY, ABRAHAM, Research Chemical Engr., General Chemical Div., Allied Chemical Corp., Morristown.

VOLZ, M. C., Mech. Engr., Herbert Fox, Cons. Engr., Newark.

New York

BAMBERGER, J. A.,* Research Dir., Brookhaven National Laboratories, Upton, Long Island.

FROEHLICH, A. O., Constr. Engr., International Business Machines Corp., Poughkeepsie.

MCMAHON, R. F., Sales Engr., Westinghouse Electric Corp., Syracuse.

PASSIGLIA, J. S., Mech. Design Draftsman, Voorhees Walker Smith Smith & Haines, Long Island City.

PORTER, R. W., Mech. Engr., U. S. Army Corp. of Engineers, N. Y. District, Construction Div., New York.

RICH, HOWARD, Estimator, W-M Metal Works, Long Island City.

SALTZGABER, R. H., Design Engr., Bausch & Lomb Co., Rochester.

SHAPIRO, MAURICE, Estimator, W-M Metal Works, Long Island City.

TAVERNA, C. S., Gen. Mgr., Taverna Fuel Co., Inc., Staten Island.

WHITE, W. N., Indus. Salesman, Wolverine Tube Div., New York.

ZUCK, A. C., Proj. Supvsr., Syska & Hennessy, Inc., New York.

REGION II

Canada

BISZTYGA, A. Z., Design Engr., Tobac Curing Systems, Ltd., Simcoe, Ont.

BOYLE, P. J., Contr. Engr., Ryan Bros. Equipment & Canadian Ice Machine Co. Ltd., Burnaby, B. C.

CHAPKIN, A. I., Cons. Mech. Engr., Toronto, Ont.

COLMAN, D. R., Proj. Engr., Canadian Comstock Co., Ltd., Leaside, Ont.

DUGAN, J. T., Sr. Contract Engr., The Powers Regulator Co., Downsview, Toronto, Ont.

GRIEVE, W. J., Sales Repr., Vampco Aluminum Products, Ltd., Strathroy, Ont.

HARPER, W. H.,* Pres., Beaver Engineering Co., Toronto, Ont.

LEUNG, A. W.,* A-C Engr., O. G. Moffat, Ltd., Hamilton, Ont.

MCADIE, L. J., Mgr., Contr. Div., H. D. Layfield & Co., Vancouver, B. C.

PALMER, DIXON, Sales Engr., Western Agencies, Ltd., Vancouver, B. C.

PELLETIER, FRASER, Repr., Aluminum Goods, Ltd., Toronto, Ont.

VAN BOEYEN, CLARENCE, Sales Mgr., Gordon Latham, Ltd., Vancouver, B. C.

WEBB, R. C., Sr. Mech. Engr., Canadian Chemical Co., Ltd., Edmonton, Alta.

REGION III

District of Columbia

MERLE, ANDRE,† Cons. Engr., Andre Merle Associates, Washington.

Pennsylvania

AMEY, A. M., Chief Mech. Engr., George M. Ewing Co., Philadelphia.

BILECKY, M. N., Sales Engr., Barber-Colman Co., Pittsburgh.

BONIDIE, C. J., Pres., Carl J. Bonidie, Inc., Bethel Park.

CHAPPLE, W. M., Vice-Pres., Chapple Bros. Inc., Johnstown.

FOOKS, J. H., Dir. of Engrg., Consumer Products, Westinghouse Electric Corp., Pittsburgh.

INGERSOLL, H. H., JR., Design Engr., C. S. Leopold Inc., Philadelphia.

KOWALSKI, S. J.,* Engr., Philadelphia Electric Co., Philadelphia.

LEBAIR, R. S., Secy.-Treas., Delaware Valley-York, Inc., Philadelphia.

MCVAY, F. P., Chief Draftsman, Patterson-Emerson Comstock, Inc., Pittsburgh.

MILLER, W. C., Vice-Pres. Engrg., Redmanson Corp., York.

SPANG, J. B., Engr. & Asst. Secy., Riggs Distler & Co., Inc., Philadelphia.

STIPE, EDWIN, III, Htg. Engr., Joseph E. Biro & Assocs., Easton.

Virginia

ALGER, R. M., Pres., Virginia Plumbing & Heating Corp., Richmond.

REGION IV

Florida

GRANT, W. J., A-C. Supvsr., General Telephone Co. of Florida, Tampa.

ZIBMAN, H. M., Chief Engr., Miami Air Conditioning Co., Miami.

Georgia

SOLOMON, H. F., Sales Engr., Thoben Elrod Co., Atlanta.

North Carolina

HAHN, R. W., Repr., Robt. E. Mason & Co., Charlotte.

JONES, B. F., Sales Engr., The Powers Regulator Co., Charlotte.

TYREN, T. T., Pres., Durham Plumbing & Heating Co., Inc., Durham.

South Carolina

BAILEY, J. H., Vice-Pres., Climatic Equipment Co., Inc., Charleston.

HOWARD, E. M., A-C. Engr., F. A. Bailey and Sons, Inc., Charleston.

MOMEIER, R. H., JR., A-C Engr., F. A. Bailey and Sons, Inc., Charleston.

OTHERSEN, C. B., Br. Mgr., Climatic Equipment Co., Columbia.

REGION V

Illinois

KING, J. A., Sales Engr., A. W. Cash Co., Decatur.

WILLIAMS, C. J., Vice-Pres. & Gen. Mgr., A. W. Cash Co., Decatur.

Indiana

DONALDSON, B. F., JR., Asst. Prod. Mgr., Whirlpool Corp., Evansville.

Ohio

PATERSON, R. W., Power Engr., The Cleveland Illuminating Co., Cleveland.

SEEBACH, H. F., Sales Engr., The Lau Blower Co., Dayton.

SMITH, R. A.,† Refr. Engr., Tibbetts Plumbing & Heating Co., Dayton.

SPEICH, C. F., Principal Mech. Engr., Battelle Memorial Institute, Columbus.

WINSTEL, F. H., Repr., Auto-Heat, Inc., Cincinnati.

REGION VI

Illinois

BOSCACCY, NATHAN, Htg. & Vtg. Design Engr., Chicago Board of Education, Bureau of Architecture, Chicago.

JORN, A. T., Mech. Engr., Abbott Laboratories, North Chicago.

LAKOSKE, RAYMOND, Mech. Designer, Chicago Park District, Chicago.

LINDBERG, R. E., Mech. Engr., Samuel R. Lewis & Assocs., Chicago.

TOMKINS, R. C., Repr., Silbrico Corp., Chicago.

Iowa

TRUNZO, N. L., Proj. Engr., Mason & Hanger-Silas Mason Co., Burlington.

Michigan

ALBERN, W. F., Appl. Engr., York-Detroit Contractors, Detroit.

DANEK, F. L., Owner, Danek Refrigeration Service, Owosso.

KRAPEK, ANTON,† Vice-Pres., E. Roger Hewitt Assocs., Inc., Lansing.

THOMAS, C. C., Mech. Engr., Commonwealth Associates, Inc., Jackson.

Wisconsin

ANDREY, GENE, Supt. of Constr., York Wagner Distributors, Milwaukee.

IGLEHART, M. M., Dvlpt. Engr., Cutler-Hammer, Inc., Milwaukee.

WEATHERSBY, W. R., Gen. Mgr., Ace Refrigeration & Heating, Milwaukee.

REGION VII

Alabama

SIMPSON, S. S., JR.,* Br. Mgr., Barber-Colman Co., Birmingham.

Missouri

BROWN, F. W., Asst. Chief Engr., Dan Truog & Clyde Nichols, Inc., Kansas City.

Tennessee

BAYER, B. M., Prof., Mech. Engrg., Vanderbilt University, Nashville.

BLAKE, J. R., Repr., Power Equipment & Engineering Co., Memphis.

POWELL, H. L., Engr., Yager & Nenon, Inc., Memphis.

REGION VIII

Oklahoma

SCHUELEIN, V. R., Jr. Engr., Carnahan & Thompson, Oklahoma City.

Texas

LOCKHART, G. H., Owner & Mgr., G. H. Lockhart Sheet Metal, Austin.

REED, H. M., Partner, Slater-Reed Co., Lubbock.

REGION IX

Utah

OLSON, D. A., Pres., D. A. Olson Co. Inc., Salt Lake City.

REGION X

Arizona

CUSHING, R. J., Draftsman, Biddle & Young Engineering, Phoenix.

HUFF, H. L., Field Engr., Tidmarsh Engineering Co., Tucson.

MILLER, D. N., Com. Sales Engr., Minneapolis - Honeywell Regulator Co., Phoenix.

SUITER, C. H., JR., Br. Mgr., The Powers Regulator Co., Phoenix.

California

EBERLE, E. M., Sales Engr., DeLaval Turbine Pacific Co., Millbrae.

GRAYSON, J. A., Owner, Burns Refrigeration, Monrovia.

JONES, M. E., Service, Scott Co., Oakland.

JORDAN, C. H., Designer, Buonaccorsi & Murray, San Francisco.

KNAPP, R. A., Supt., Jackson & Blanc, San Diego.

MITCHELL, G. L., Sales Engr., Scott Co., Oakland.

PALMER, J. A., Repr., California Hydronics Corp., San Francisco.

TOMBLIN, DAVID, Designer, Hughes Aircraft Co., Culver City.

WOLTERS, C. E.,* Supvsg. Mech. Engr., Buonaccorsi & Murray, San Francisco.

FOREIGN

England

BONNER, SIDNEY, Chief Engr., Robertson E. Woodcock, Ltd., Woodford Green.

JACKMAN, R. W., Chief Refr. Engr., Acrow (Engineers) Ltd., Saffron Walden.

India

D'SOUZA, WILLIAM, Prop., General Refrigeration Engineering Works, Mangalore, Mysore.

Iran

SHOKRAIN, A. A., Gen. Mgr., Sapphire Engineering Co., Tehran.

Venezuela

THUMANN, J. F., Sales Engr., Liquid Carbonic Venezolana S.A., Caracas.

Vietnam

VAN LAM, TRUONG, Refr. Engr., Vietnam Development Co., Saigon.

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EVAPORATIVE COOLING

(*Continued from page 54*)

Air is circulated through the stacks and rows of field boxes at an outlet velocity from the cooler of 500 to 750 fpm. The canvas baffle at the evaporative cooler discharge grille prevents short circuiting of the air. Drop curtains are used to seal off the ends of each sweat room.

Ethylene gas is introduced by the trickle method. It is delivered through a small copper tube in a continuous flow of about five parts per million parts of air by using suitable reducing valves attached to the high pressure cylinder in which the gas is purchased. Generally, the fruit degreens in 72 to 96 hrs.[20] To avoid excessive deterioration of the fruit during the sweating operation constant and careful supervision is required. After sweating, the fruit should be thoroughly ventilated and kept cool.

The calculations of cooling down and degreening citrus fruit are illustrated in the following example. (Numbers are rounded off in the computations.)

GIVEN

a. 20 x 40-ft sweat room with 10-ft ceiling and two exposed walls. Heat transfer coefficient of insulated ceiling 0.10 and of masonry wall 0.30.

b. Fruit capacity of sweat room 1600 50-lb field boxes of oranges. Weight of boxes 12 lb.

c. Specific heat of oranges 0.90. Specific heat of field boxes 0.40.

d. Heat of evolution of oranges 8000 Btu per ton per 24 hr at 80 F and 6500 Btu at 70 F.

e. Humidifying effectiveness of evaporative cooler 80%. Maximum air delivery 6500 cfm.

f. Design outside dry bulb temperature of 85 F. (24 hr aver.)

g. Design outside wet bulb temperature 63 F.

h. Design inside dry bulb temperature 70 F.

i. Design inside wet bulb temperature 68 F.

j. Outside air with weight of 13.9 cu ft per lb of dry air having a specific heat of 0.24.

COOLING DOWN CALCULATIONS – 24 HRS

1. Heat Load (sensible)

Walls: $60 \times 10 =$

$$600 \times 0.30 \times 15 = 2700 \text{ Btu}$$

Ceiling: $20 \times 40 =$

$$800 \times 0.10 \times 15 = 1200$$

Fruit: $1600 \times 50 =$

$$\frac{80000 \times 0.90 \times 15}{24} = 45000$$

Boxes: $1600 \times 12 =$

$$\frac{19200 \times 0.40 \times 15}{24} = 4800$$

Respiration:

$$\frac{80000 \times 8000}{24 \times 2000} = 13300$$

Total 67000

2. Cooler outlet temp $= 85 - [(85 - 63) \times 0.80] = 67$ F

3. Minimum temp rise possible

$$= \frac{67000 \times 13.9}{0.24 \times 6500 \times 60} = 10 \text{ F}$$

INTER-SOCIETY COMMITTEES
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ASHRAE OFFICERS, COMMITTEES
See page 70, this issue

STANDARDS COMMITTEE
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TECHNICAL COMMITTEES
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RESEARCH AND TECHNICAL COMMITTEES
See page 84, this issue

4. With 67 F air supply and a temperature rise of 10 F using 100 % fresh air maximum air delivery the final conditions in the sweat room would be 77 F and 58 % relative humidity. Additional free moisture from the spray nozzles in the evaporative cooler is required. The evaporation of this moisture will cool the room by the latent heat required to vaporize the moisture. The skeleton psychrometric chart in Fig. 7 illustrates the process whereby the temperature in sweat room is brought down to 70 F with 90 % relative humidity.

5. Fresh air $=$

$$\frac{67000 \times 13.9}{0.24 \times 15 \times 60} = 4300 \text{ cfm}$$

6. Moisture $=$

$$\frac{(0.0143 - 0.0114) \times 4300 \times 60}{13.9} = 50 \text{ lb}$$

DEGREENING CALCULATIONS

1. Heat load (sensible)

Walls and ceiling:

$$2700 + 1200 = 3900 \text{ Btu}$$

Respiration:

$$\frac{80000 \times 6500}{24 \times 2000} = 10800$$

2. Fresh Air $=$

$$\frac{14700 \times 13.9}{0.24 \times 15 \times 60} = 950 \text{ cfm}$$

3. Moisture $=$

$$\frac{(0.0143 - 0.0144) \times 950 \times 60}{13.9} = 12 \text{ lb per hr}$$

Since the air delivery of the evaporative cooler using both sprays and fan is 6500 cfm, the mixing damper and exhaust air vents should be adjusted to provide 4300 cfm fresh air during the cooling down period, and 950 cfm fresh air during the degreening operation.

Another use for evaporative cooling is as a supplement to refrigeration in the storage of citrus fruit. Although oranges have a long harvesting season and are considered a fresh fruit crop, marketing conditions and other factors may make it advisable to hold them in temporary storage at the packing house for as long as one month. And in the case of lemons and grapefruit these crops are usually stored for some time. A large portion of the former crop is picked during the period of least consumption.[16] Recommended storage conditions for oranges, lemons and grapefruit are as follows.[20]

	Oranges	Lemons	Grapefruit
Temperature	50	58	60
Relative humidity	88-90	84-88	86-88

While these conditions require refrigeration in the summer they can often be met with evaporative cooling during the fall, winter and spring when the outside wet bulb temperature is low.

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BETTER COMPRESSORS

(*Continued from page 59*)

one or more cylinders are mechanically held open. By holding open the suction valves, the suction gas is drawn into the cylinder and, without being compressed, returned to the suction line. The use of this system allows for great flexibility.

Single step unloading, where usually 50% of the compressor is unloaded, may be employed, or multi-step unloading, resulting in capacity reduction in steps of 25% or 33-1/3%, depending on the number of cylinders in the controlled compressor. One other advantage of this system is that the power reduction is almost directly proportional to the rate of unloading, plus the fact that the frequent starting and stopping of the unit is reduced greatly.

In cylinder unloading type capacity controls, the power for holding the suction valves open may be either the oil pressure from the lubrication system or discharge gas pressure. The hydraulic system is generally totally within the crankcase, whereas the control on the hot gas arrangement is external. Both arrangements are being used successfully, with the hydraulic one having the advantage of being easily adapted to pneumatic control from a central control system.

Starting – To start a compressor under load, motors with starting torque of 225% are generally used. To reduce the inrush current at start-up, and in order to use standard torque motors, the compressor is quite often equipped with an arrangement which renders all cylinders inoperative until such time as the motor has obtained its full speed. This is referred to as no load start. A true no load start condition usually exists with the hydraulic unloading system, since

when there is no oil pressure the cylinders are unloaded and vice-versa. On the hot-gas unloading system, no pressure differential exists (unless all valves are absolutely tight), and, therefore, the compressor must have a few turns to create a pressure differential.

In talking about starting a compressor, it might be wise to mention a common error made in selecting motor hp, especially on commercial and low temperature installations.

It is known that at a reduced suction pressure, a lower motor hp is required to operate at a given condition. Quite often this brake hp is taken and the nearest motor hp selected. This can lead to serious problems on start-up when the suction pressure is high. Some means, such as suction pressure regulator or limitation of compressor capacity, must be employed if the unit is to remain on the line.

What are the measures of a good design-engineering and manufacturing job on a compressor? Certainly, they would include low clearance volume, little vibration, low noise level, leak-proof, and consistent performance.

INSULATION SYSTEM

(*Continued from page* 62)

tion, the rate of which for the new system is about one-tenth that for the present system.

Thus, the use of the polyester-acrylic insulation system significantly lessens the disadvantages attendant to the use of paper-enamel-cotton systems. Its lower moisture content and excellent moisture removal characteristics should allow a reduction in compressor dehydration cycles, providing increased capacity from existing dryout facilities. Its greater thermal stability allows the use of higher dehydration temperatures for still shorter drying times and permits more flexibility in selecting suitable locked, motor, overload protection. The thermal and chemical stability of this new insulation system may eliminate the need for special compressor cooling devices as well as aid the trend toward increased capacity in smaller size units.

PARTS AND PRODUCTS

PORTABLE AIR CLEANER

Light enough to be moved easily from room to room, this plug-in electronic air cleaner has a built-in, two-speed motorized fan, and requires no water or drain connection. The "Room Size" model has an air handling capacity of 225 cfm at 90% efficiency, as determined by the National Bureau of Standards Dust Spot Test.

Electro-Air Cleaner Company, Inc., Dept. RS, Olivia and Sproul Streets, McKees Rocks, Pa.

COMPACT FURNACES

Gas and oil-fired "MY" series furnaces in a space-saving design are available in counterflow, up-flow, basement and horizontal models. Capacity of gas models is AGA-rated at 105,000 Btu/hr input, oil-fired at 100,000 Btu/hr output.

All furnaces of this series can be equipped for summer cooling. For this function, an evaporator control is placed in a casing and installed in the system. Condensers are the remote type.

Stewart-Warner Corporation, Heating and Air Conditioning Div, Lebanon, Ind.

ELECTRONIC RACK COOLER

Requiring only 12¼ in. of rack height and mounting in standard 19 in. construction, this self-contained miniature air conditioning unit is designed to maintain temperature control of rack mounted electronic equipment. Features of the Model RC-6 cooler include permanently lubricated bearings, automatic temperature control, and sealed refrigeration system.

Airflow Company, 400 South Stonestreet, Ave., Rockville, Md.

TEST CABINETS

Drop from room temperatures to −110 F may be achieved in less than five min by using liquid carbon dioxide sprayed into the chamber through a special nozzle. No compressor is needed; the liquid chemical is ejected directly from the nozzle.

The Hi-Lo Frigid-Cab is 6 x 11 cu ft. In temperature ranges from −100 F to room temperature and 100 to 300 F. It is engineered to operate within plus or minus 2 deg accuracy.

Insulated throughout with 6 in. of special glass fiber, the unit features simplified controls to regulate and indicate temperature, and a circulating fan to eliminate stratification, assuring top to bottom temperature uniformity.

Heating is achieved with a Calrod, fin-type stainless heater.

Hudson Bay Company, 3070-82 W. Grand Ave., Chicago 22, Ill.

SILICONE ADDITIVE

Glass or mineral wool insulation is cited as having greater resiliency, heat stability and resistance to the damaging effects of moisture when this amino-functional silane, Z-6020, is added to the binder resins.

This additive contains primary and secondary amine groups, and will combine with the organic resins commonly used as insulation binders. Since it hydrolyzes in water to produce hydroxyl groups, it will also attach to inorganic materials as glass fibers or mineral wool. The material

is considered a coupling agent between the inorganic insulation and the resinous binder.

Water resistance tendencies of the material lead to greater wet strength of the insulation.

Dow Corning Corporation, Midland, Mich.

CONDENSING UNIT

Use of a permanent split capacitor motor which eliminates the starting relay and capacitor, and an oversized coil large enough to store refrigerant, thus eliminating a liquid receiver, are features of this RC-P-2 hp condensing unit. Design of the condenser, which has a cooling capacity of 24,000 Btu/hr and a motor of 230-volt single phase only, allows for either original installation or integration with existing heating systems, and for installation with either a capillary tube or thermal expansion system.

Stewart-Warner Corporation, Heating and Air Conditioning Div, Lebanon, Ind.

EPOXY IMPREGNATED COILS

Coils encapsulated in epoxy resin offer maximum protection against moisture, water, machine oils, diluted acids and harmful alkalies. The encapsulation equipment replaces protection previously given by cotton tape and varnish impregnation insulation.

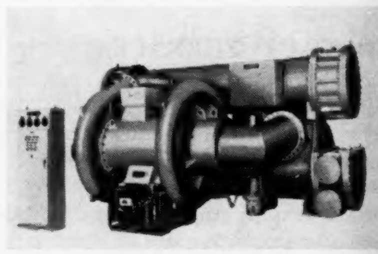
Two different processes are used; one type is fully automatic and the other embodies a non-automatic, batch system. In both cases, the coils are impregnated with a heavy shock-resistant, non-flammable layer of epoxy.

Commercial and industrial solenoids, water valves and hot gas defrost valves are among the products now offered standard with the encapsulated coils.

Controls Company of America, Schiller Park, Ill.

INDUCTOR AIR CONDITIONERS

Low pressure inductor units for application to perimeter areas of multiroom buildings using conventional



ductwork are in five unit sizes: 20, 28, 36, 48 and 60 in. (tube length of the secondary coil). Up to 700 cfm of fresh air may be supplied by each low pressure inductor air conditioner.

Individual room control of primary air is provided by a standard indexing and locking (field set) device for fixing unit volume. It may also be manually operated through a remote control arrangement if desired. The secondary coil, either hot water or steam, may be controlled manually by a hand valve from the standard access door.

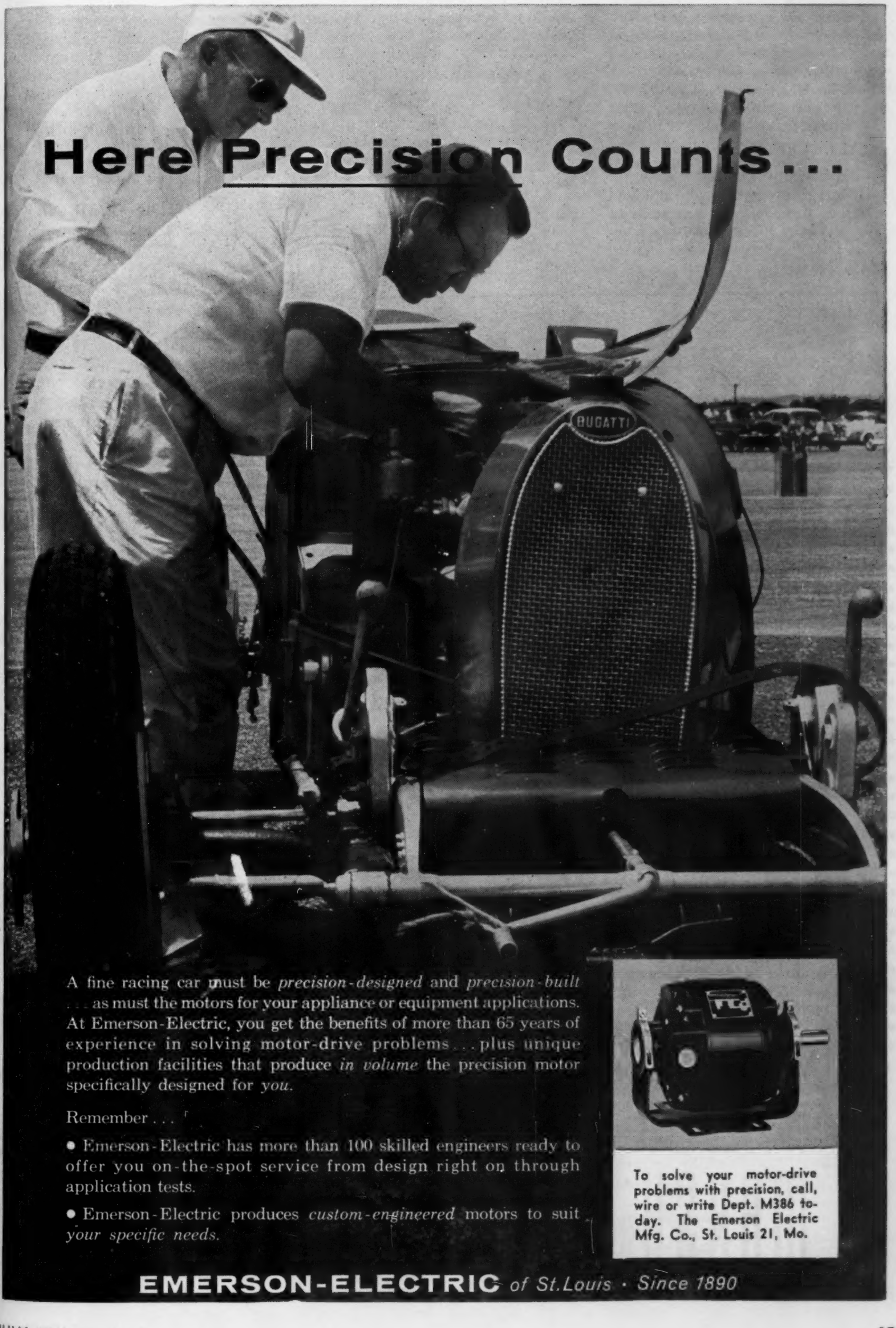
Low pressure inductor coils are with four, five or six tubes in face, single row, in hot water and steam construction.

American Radiator and Standard Sanitary Corporation, Industrial Div, Detroit 32, Mich.

VAPOR BARRIER

Foil and kraft paper bonded together with a flame-extinguishing adhesive comprise this vapor barrier for pipe jacketing and duct insulation facing.

When the temperature surrounding this material reaches the combustion stage, gases or vapors are released which tend to smother the flame. Identified as Pyro-Kure, it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does not depend for its fire-resistant properties on corrosive salts which may leach out and entirely dissipate the flame resistance. It may be used with either hot or cold piping.
American Sisalkraft Corporation, 55 Starkey Ave., Attleboro, Mass.

SMALL FREEZER

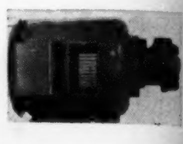
To the present line of 20, 17 and 12 cu ft upright freezers has been added a 10.4 cu ft model, with a capacity of 365 lb. Model FC-10V features magnetic door gaskets, a safety device which permits door opening with a slight outside pull or inside push.
Whirlpool Corporation, St. Joseph, Mich.

ISOLATOR MOUNTINGS

All-rubber housings are a feature of Type LR machinery mountings with steel springs as the vibration and shock absorbing medium. Rubber, with low acoustical impedance, prevents the transmission of structure-borne high-frequency disturbance and noise.

Intended for use with fans, cooling towers, furnaces, piping, packaged air conditioners, air handling units, turbines and motor generator sets, it may carry loads ranging from 150 to 1400 lb per isolator. Minimum operating height is 4 in. Opposed five slots in the base of the unit permit it to be removed from an installation by rotating the amount 45 deg.
Korfund Company, Inc., 48-53D 32nd Place, Long Island City 1, N. Y.

PUMP-MOTOR COMBINATION



Developed specifically for applications where space and weight are at a premium, this Series 42 pump and motor combination measure 9⅛ x 5 in. It incorporates a totally enclosed electric motor rated for 1/12 to 1/6 hp, and an internal gear pump with a capacity of from 1/3 to 3 gpm. Twenty-four models are in the series. All may be furnished either with or without a built-in relief valve.
Tuthill Pump Company, 965 E. 95th St., Chicago, Ill.

MULTI-DUTY CHECK VALVE

Constructed of brass throughout, with Teflon seating and Inconel-x springs, this check valve will function at high or low pressure, high or low flow, extremes of temperature, and with gas or liquid. Its durability and freedom from reactions with all synthetic refrigerants and oils are said to insure long life.
Chatleff Valve and Manufacturing Company, P. O. Box 996, Austin, Tex.

PROPELLER FAN

Four-bladed and made of acrylonitrile copolymer plastics material, this propeller fan is designed for high volume production of air-moving devices such as portable appliances and electronic cabinet cooling units. The 4-in. diam plastics fan rotates clockwise, as high as 11,000 rpm. It is for a unit bearing mount.
Torrington Manufacturing Company, Air Impeller Div, Torrington, Conn.

SPACE SAVING MOTOR

For such applications as machine tools, fans and blowers this motor offers more hp in less space. Identified as the Thinline, it reduces motor overhang, takes up less aisle space than standard flange-mounted motors, and allows more compact design of equipment such as ventilating fans.

It may be mounted in either a horizontal or vertical position, and is

(*Continued on page 105*)

BULLETINS

(Continued from page 75)

recting acidity and alkalinity, and a chart indicating proper amounts of calcium chloride needed in preparing or strengthening brine.
Calcium Chloride Institute, 909 Ring Building, Washington 6, D. C.

Capacitor Selection Guide. Tables for choosing correct ratings of switched shunt capacitors for modern induction motors cover all standard types, enclosures, and nominal speeds, 220 to 4000 volt, 2 to 500 hp. GED-3687, 6 pages.
General Electric Company, Schenectady 5, N. Y.

Industrial Compressors. Heavy duty four-cylinder compressors for refrigeration service are illustrated and graphically described in Bulletin 651-C.
Frick Company, Waynesboro, Pa.

Vinyl Coated Steel. How one manufacturer of air conditioning units incorporated the recently developed vinyl coated steel in its line of portable air coolers and heaters is described in 4-page Bulletin No. 7.
United States Steel Corporation, 525 William Penn Place, Pittsburgh 30, Pa.

Cleaning of Refrigeration Systems. Filtering and cleaning of refrigeration systems is detailed in Folder A-12, outlining a six point procedure for cleaning after hermetic burn-outs before installing new or re-built motor compressor. It also contains data on low side filtration, plus dimensional information and ratings of the complete line of Permaclean Filters.
McIntire Company, Livingston, N. J.

Flexible Urethanes. Basic physical and chemical data on both polyester and polyether types of Mopcofoam, flexible urethanes is contained in this 16-page booklet.
Nopco Chemical Company, Plastics Div, 175 Schuyler Ave., North Arlington, N. J.

Recorders and Recording Controllers. Continuous self-standardization without dry cell, standard cell or standardization mechanism, and electronic servo-operated measuring systems affording high accuracy while providing sufficient torque to operate a variety of control devices, characterize the group of strip-chart recorders and recording controllers for measurement of electrical and process variables described in 12-page Bulletin GEA-6887.
General Electric Company, Schenectady 5, N. Y.

Insulated Building Panels. Consisting of a rigid plastics core, firmly bonded and sandwiched between aluminum sheets or aluminum in combination with other materials, Alply is offered as providing notable simplification in certain architectural and allied applications. Available in a wide range of strength, insulation value, surface finish and color, it can be cut, mitered, shaped and formed. Included in full color, 24-page, descriptive Form 70-11265 is a section on technical data, describing the panel's thermal, acoustical, corrosion-resistance and strength characteristics.
Aluminum Company of America, 779 Alcoa Building, Pittsburgh 19, Pa.

Pipe Welding. Two technical articles on small pipe welding, written by A. N. Kugler, chief welding engineer with this firm, are descriptive of cor-

rect practices. They are entitled, "Fabrication of Small Piping by Welding and Brazing," and "Designing for Welded Systems of Small Size Pipe."

The first defines and describes the welding processes applicable to the joining of small pipe; it covers applied welding engineering, fields of application, welding processes and equipment, and brazing techniques.

The second covers the principles of welding and brazing design and how these may be applied to the various pipe metals. (ADR 118)

Air Reduction Sales Company, 150 E. 42nd St., New York 17, N. Y.

Smoke Alarm Control. Bulletin 553 is on the Electric Eye unit, which gives split second alarm or will shut down circulating fans when smoke enters the air conditioning system from any number of sources.

Photomaton, Inc., 96 S. Washington Ave., Bergenfield, N.J.

Aluminum Welding. "Aircomatic and Heliwelding of Aluminum" is the subject of a 120-page spiral-bound presentation of techniques of welding aluminum using the gas-shielded metal-arc and tungsten-inert-gas processes. Major topics covered include weldability of aluminum and its alloys, definitions and technical discussions of the two welding processes, selection of the proper process for certain job applications, descriptions of manual and automatic equipment (with photographs, charts and detailed diagrams), welding power supplies, and safety practices. Specify Form ADI 1258.

Air Reduction Company, Inc., 150 East 42nd Street, New York 17, N. Y.

Blower Performance. Two technical data sheets on the performance and dimensions of the Radiax blower units describe materials and finishes, list important dimensions and give pressure, speed, power and current curves for each blower unit size. Bulletin RX1 covers fan unit sizes up to a maximum outside dimension of 17 in.; Bulletin RX2 includes two unit sizes with maximum dimensions of 20 and 24 in. respectively.

Torrington Manufacturing Company, Air Impeller Div, Torrington, Conn.

Multi-Pointer Indicators. Illustrating the group of individual miniature vertical gage units known as Bailey Mini-Line Multi-Point Indicators, 4-page Product Specification P11-1 provides details of pointer movement, 3¼-in. scale design and optional internal illumination.

Bailey Meter Company, 1050 Ivanhoe Road, Cleveland 10, Ohio.

Process Moisture Monitor. Pointing up the operation, applications and specifications of this instrument for continuous monitoring or control of water in gaseous streams is 4-page illustrated Bulletin 1845.

Consolidated Electrodynamics Corporation, 360 Sierra Madre Villa, Pasadena, Calif.

Inherent Protection For Three-Phase Motors. Illustrated flyer GEA-6932 includes description of product features, motor ratings, and connection diagrams for U.L.-listed inherent protection for integral-hp, three-phase motors for fan, blower, and compressor applications.

General Electric Company, Schenectady 5, N. Y.

Maximum Capacity Ball Bearing. For applications demanding sustained performance under specially heavy radial loads, the Super Max bearing, as described in Bulletin 107, utilizes the maximum number of large-size balls that may be safely inserted in the

(Continued on page 103)

Others are saying—

(Continued from page 71)

radiated by a cooling tower is determined almost entirely by fan noise, an estimate of cooling tower noise may be obtained with knowledge only of the fan horsepower. *Noise Control, May 1959, p 44.*

that . . . . . . if any of the variables associated with fan operation are constant for a specific problem, then a grouping containing this "constant variable" may be used for studying the interaction of the remaining variables. Fan Laws Simplified is the title of the article. *Air Conditioning, Heating and Ventilating, May 1959, p 84.*

that . . . . . . the use of the capillary tube as a refrigerant control method has a disadvantage in the fixed resistance to flow of the capillary, which causes its capacity to be dependent upon the physical properties of the liquid refrigerant and the pressure difference across it. It is concluded that a longer, larger diameter capillary tube gives a more uniform flow over a wide range of operating pressures than a shorter, smaller tube. *Refrigeration Journal, April 1959, p 51. (Australian)*

that . . . . . . previously published methods for the solution of one-dimensional heat-conduction problems with melting or freezing are briefly reviewed and weaknesses of previous analytic and numerical methods are outlined in this paper. Two new and more generally useful numerical methods, applicable to digital and analog computation, are developed here. *Transactions of the ASME, May 1959, p 106.*

that . . . . . . careful selection, as to size and type, of those potatoes used in preparing frozen French-fries for packaged sale may affect the success or failure of their marketing. This installation uses a blast-type tunnel, cooled to a temperature of —30 F, in which the potatoes, pre-cooled before being frozen, remain for thirty minutes. *Canner/Packer, June 1959, p 32.*

that . . . . . . a well designed kitchen ventilating system an essential for air conditioned food service establishments must include sizing of the exhaust fans to provide an air velocity of 50-100 fpm across the face of the hood, or 20 to 30 air changes per hr in the kitchen, whichever is greater. Unhooded cooking appliances add tremendously to the cooling load, and should never be installed if there is any possible way to locate them under an exhaust hood. Basic factors are covered that determine the required fan capacity. *Air Conditioning, Heating and Ventilating, April 1959, p 89.*

BULLETINS

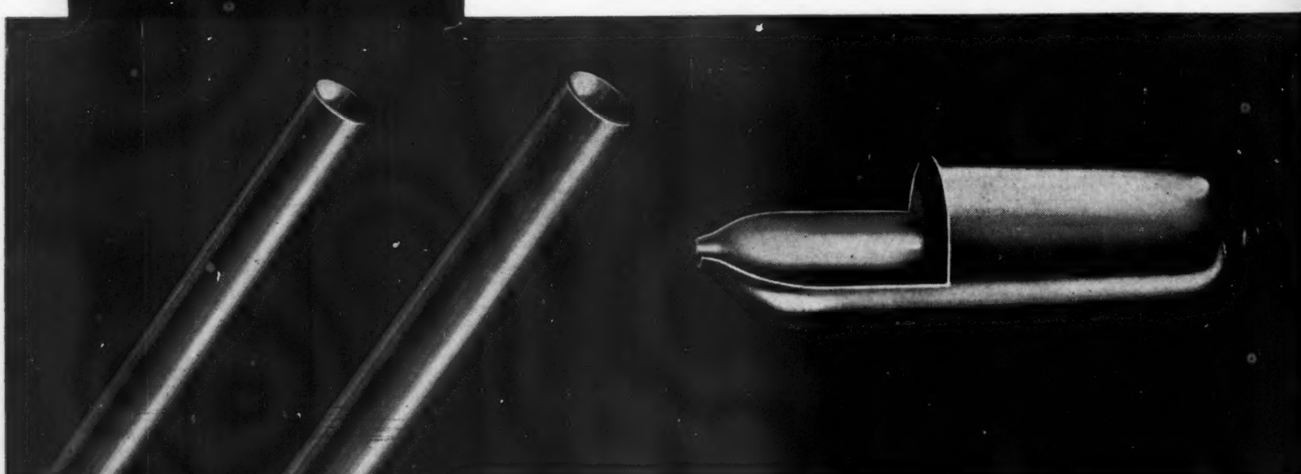
(Continued from page 100)

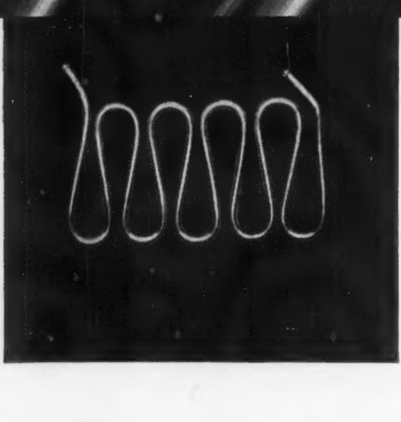
bearing by way of an accurately placed filling slot. The design of the filling slot is such that the bearing will also carry combined radial and thrust loads where some thrust load-carrying ability is essential.

The bearings are in light, medium and heavy series, and are open or with single or double shields.
Hoover Ball and Bearing Company, 5400 South State Rd., Ann Arbor, Mich.

Power Transmission Machinery. Featuring Flexidyne Dry Fluid Drives and Para-flex Flexible Cushion Couplings, 8-page Bulletin A-706 briefly reviews this manufacturer's line of power transmission machinery, offering illustrated descriptions of steel conveyor pulleys, roller chain drives, various types of shaft couplings, V-belt drives, and other products.
Dodge Manufacturing Corporation, Mishawaka, Ind.

Non-Woven Material. Filtration, insulation, sound damping, padding and abrasion resistance are various applications of Troyfelt, the synthetic non-woven material described in 8-page Bulletin 101-559-5M. Detailed are areas such as strength, dimensional stability and chemical resistance.
Troy Blanket Mills, 200 Madison Avenue, New York 16, N. Y.





PARTS AND PRODUCTS

(Continued from page 96)

in polyphase dripproof and enclosed construction from 1 to 5 hp at 1800 rpm.

Size and weight reduction of the motor is accomplished by using formed end turn windings. An insulating end punching separates the formed end turn from the stator core.

General Electric Company, Schenectady 5, N. Y.

ROOF-TOP UNIT

Heating with gas, and cooling with electricity, these packaged air conditioners may be installed on the roof of one-story buildings. They are in two sizes: 200,000 Btu/hr input heating coupled with 7½ hp cooling; and 250,000 Btu/hr heating linked with 10 hp cooling.

Each unit is set over an opening in the ceiling and fitted to a diffuser. The unit draws return air up through the center of the diffuser and discharges conditioned air down and out the sides. No ductwork is required.

Carrier Corporation, Unitary Equipment Div, Syracuse 1, N. Y.

FORCE AMPLIFIER

For the amplification of small forces, Model 301 can be actuated directly by mechanical flow, pressure or temperature measuring elements, or similar motion producing devices, requiring less than one gram (0.04 oz) force to position the amplifier, and with an output force up to 500 gram (18 oz). Output motion is linear and reproduces the input motion within 0.001 in. with a constant load.

Operating in an oil bath to provide long life for moving parts, the amplifier must be operated in a horizontal position, as the lower cover acts as an oil reservoir. Universal mounting brackets are furnished for bottom or rear mounting.

American Meter Company, Inc., P. O. Box 306, Garland, Tex.

10.2 CU FT REFRIGERATORS

Flush hinge door mountings permitting opening of the door within the width of the 26 in. wide refrigerator cabinet offer two advantages for these models: installation of the large 10.2 cu ft units in space normally required for smaller size refrigerators, and installation in corners or within ½ in. of wall cabinets. Both models, F-10S and F-10, are 59 in. high and 31⅛ in. deep, and contain a 60-lb capacity freezer section with side-hinged door designed to remain open without being held.

Whirlpool Corporation, St. Joseph, Mich.

DIGITAL TRANSDUCERS

Available in vacuum and pressure ranges to 10,000 psi, temperature ranges to 600 F, and in mercury or bellows type manometers for liquid level or flow applications, the Series 400 American Digital Transducers measure air conditioning and heating power plant variables and provide digital encoding for remote reading of flow, liquid level, pressure or temperature. Both linear and square root encoders are available.

Conservatively rated with an accuracy of ±1.0% of span, reproducibility is better than ±0.5% of span.

American Meter Company, Inc., P. O. Box 306, Garland, Tex.

CONDENSER FOR TRUCKS

Adaptability to a greater variety of body styles and makes is claimed for the redesigned Crest Condensing Unit, which is manufactured for use

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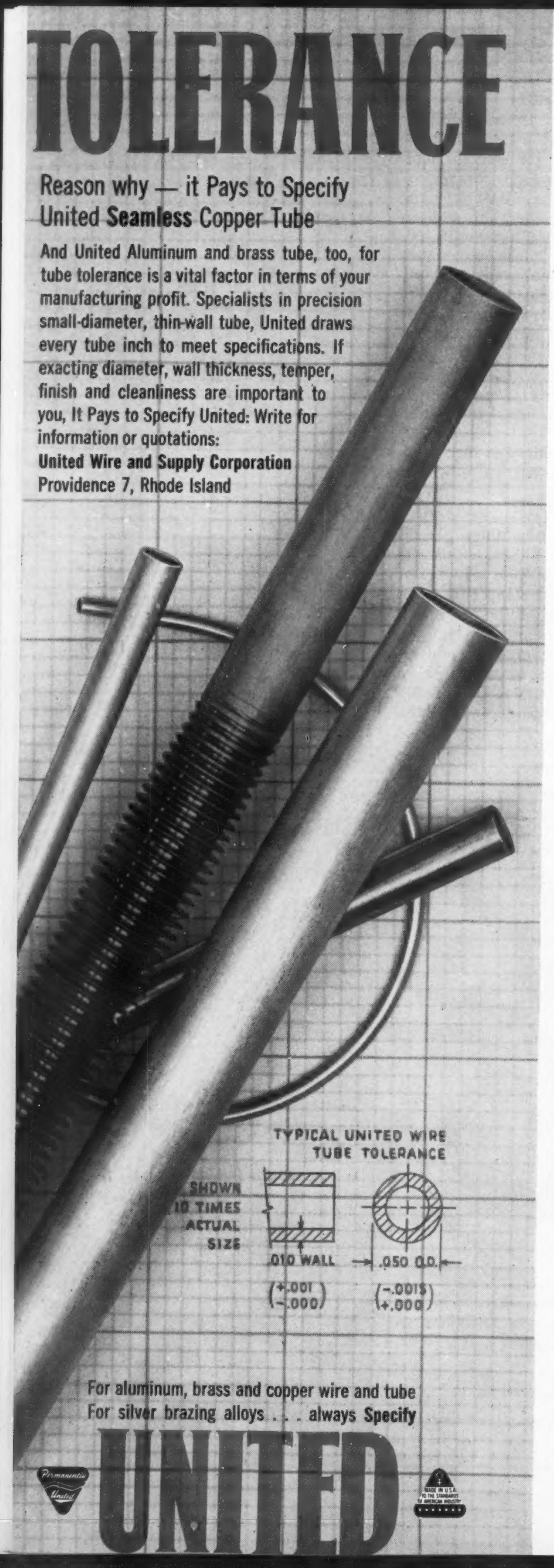

With ASHRAE Chapters

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Treas.—Daniel C. Raasch

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PARTS AND PRODUCTS

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in parked trucks. A line of five models, of ¾, 1, 1½, 2 and 3 hp, permits selection of the proper sized unit for the specific truck to be refrigerated. All models are available with three-phase or single phase electric motors.

Units of 1½, 2 and 3 hp are designed for mounting on the truck chassis rail or over the cab; the ¾ and 1 hp models are for inside mounting and are shipped without cover or mounting brackets.

Kold-Hold Div, Tranter Manufacturing, Inc., Lansing 9, Mich.

AUTOMATIC ICE MACHINE

Making up to 100 lb of crushed ice daily, this small-size Super Flaker Automatic Ice Machine may be used either as a built-in unit or as a floor-mounted machine. It requires 3 sq ft of floor space; actual dimensions are 3 x 2½ ft.

The system includes a ⅛ hp, 110-115 volt, single-phase, 60-cycle compressor using Refrigerant-12. The unit is hermetically-sealed and air-cooled.

The heavily insulated stainless steel compartment stores up to 35 lb of flake ice. Once the machine is turned on, it provides a continuous flow of flake ice into the storage bin.

Scotsman, Queen Products Div, King-Seeley Corporation, Albert Lea, Minn.

FLOOR UNIT AIR CONDITIONER

Called the Roomette, this one hp unit is recommended for use in small homes, summer cottages, and garage apartments. Installed over an opening in the floor in structures with an open crawl space or ventilated area below, the unit frees window space, and can double as an end table or night stand. Where this air space is not available, the unit is mounted directly through the wall. It has an ARI certified rating of 9,700 Btu and is thermostatically controlled.

Carrier Corporation, Syracuse 1, N.Y.

HORIZONTAL CONDITIONER

Designed for concealed overhead installations, such as dropped ceilings or closets, the Type 30 Remotaire is a horizontal fan-coil air conditioning unit for use with hot or chilled water. Sweat protection is afforded by the drain pan design, which eliminates metal-to-metal contact between the pan and the bottom of the coil. The units are available with factory sub-assembled control packages, and have either right- or left-hand piping connections.

American-Standard, 40 West 40th Street, New York 18, N. Y.

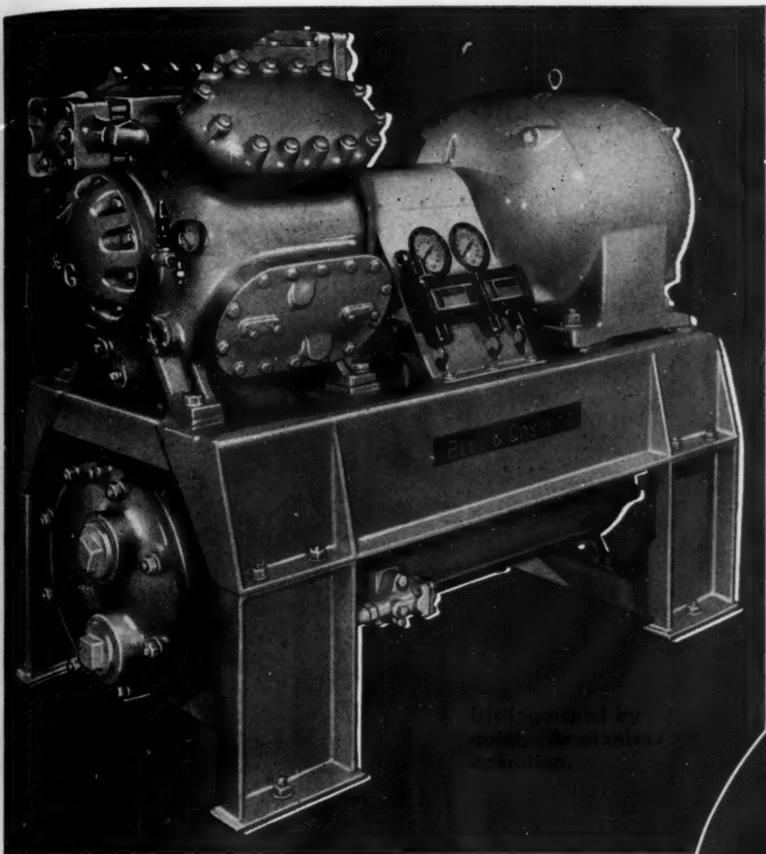
INCREMENTAL CONDITIONERS

Four-season air conditioning is cited as the aim of this new line of all-electric incremental conditioners providing both heating and cooling for either existing or projected multi-room buildings. Any unit may be installed initially for heating only.

Controls consist of an on-off switch and a combined heating-cooling thermostat. Ventilation is provided by a high pressure centrifugal blower through an automatically-controlled electrically-operated air flow damper.

There are three cooling capacities from 9000 to 15,000 Btu/hr and three heating capacities from 8000 to 15,300 Btu/hr.

Remington Air Conditioning Div, Auburn, N. Y.



AN ACHIEVEMENT IN COMPACT, *QUIET* EFFICIENCY... WITH ALL MAJOR PARTS MADE AND GUARANTEED BY BELL & GOSSETT COMPANY

Cut-away view of Condenser Compressor

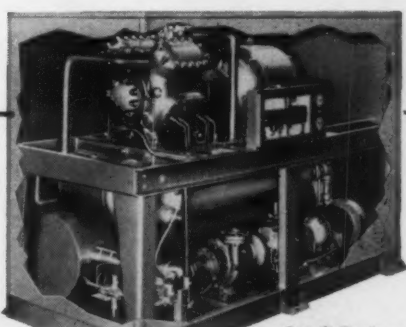
B&G CONDENSING UNIT

There are numerous, well-defined reasons why this B&G Condensing Unit is the outstanding buy in air conditioning and refrigeration equipment. An impressive list of features, many of them exclusive, are engineered into the shortest unit on the market today—with capacities ranging from 7½ through 150 tons.

Not the least of the reasons is the fact that all major components are built in the B&G plant—*hence covered by a single manufacturer's guarantee!*

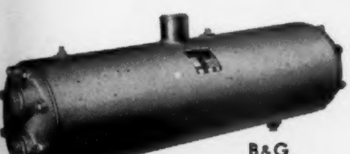
The B&G Package Liquid Cooler is another matched unit—a completely integrated and assembled water chiller. In capacities from 7½ through 150 tons, the components of this chiller give a complete capacity-engineered combination of chassis, motor, compressor and automatic control obtainable nowhere else. Everything included—no extras to buy.

Send for descriptive literature and engineering data.



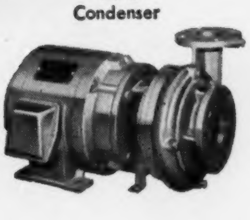
B & G Package Liquid Cooler

A FULL LINE OF REFRIGERATION AND AIR CONDITIONING COMPONENTS



B&G Condenser

B&G Evaporator



B&G Centrifugal Pumps

Reg. U.S. Pat. Off.

BELL & GOSSETT COMPANY

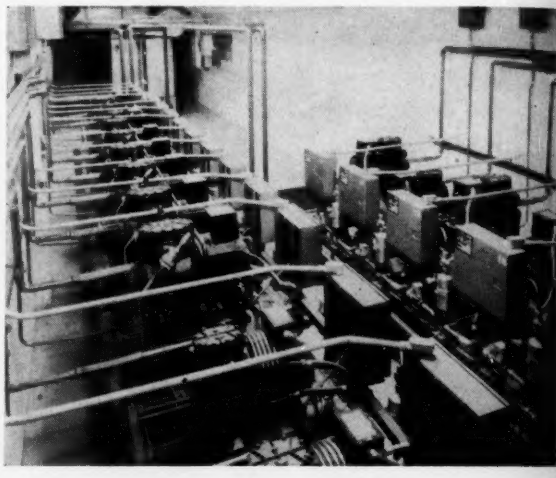
Dept. FU-64, Morton Grove, Illinois

Canadian Licensee: S. A. Armstrong, Ltd., 1400 O'Connor Drive, Toronto 16, Ontario

Applications

SUPER MARKET CLOSED WATER SYSTEM

Cooling water in a closed circuit is pumped through the evaporative water cooler tube bundle and then through all of the individual unit condensers and compressor water jackets in this Colorado Super



Market installation. Twenty-one Copelametic water cooled units, pulling an over-all heat load of 680,000 Btu/hr are connected to the Recold Corporation Dri-Fan evaporative water cooler to maintain proper case temperatures in the King Soopers Market in Denver.

Pressure-operated water valves are installed on the individual condenser outlets to provide the precise amount of water needed.

Water in the closed circuit is never changed, eliminating the possibility of scale formation in the condenser tubes and water jackets. Sizing of individual condenser circuits with the difficulty of matching circuits to their compressors is also eliminated.

INDIANA TERMINAL GARAGE HEATED WITH RADIANT SYSTEM

Seventy degree temperatures are maintained for winter heating in the Terminal Transport Company's Indianapolis garage by a system using gas-fired radiant heat generators hung from the roof beams 15 ft above the concrete floor.

The structure itself is of corrugated steel with a quonset-type roof. It is insulated, but there are many cracks and air crevices. Overhead truck doors, three on either side, are frequently opened.

Perfection Industries, a division of Hupp Corporation, supplied nine generators to radiate over the 3648 sq ft of floor area, and two small units in the washroom and office. Propane is supplied from a 1000 lb tank located outside the garage because of local restrictions.

The special nature of garage work makes it necessary to keep the floor of the area 2 deg warmer than air near the roof.